Heating·Piping and Air Conditioning

Volume 6, No. 9

SEPTEMBER, 1934

MAINTENANCE *of the* Air Conditioning Plant

The heart of an air conditioning plant. Note the portion of the circular marine type glass inspection port showing just to the left of the access door. These are located behind the sprays so that a clogged nozzle can be detected instantly

What Should be Checked and Serviced *at this time of the year?*

TIMELY AND PRACTICAL is this article, which presents suggestions for the main items of an air conditioning plant which should be checked and serviced during the autumn months, explains the reasoning behind the recommendations. While every plant differs in details, most of the items covered will be found of interest and value by all concerned with operation of an air conditioning system.

By **A. Warren Canney***

PROPER operation of an air conditioning plant is just as important as careful design. While this should be obvious, an inspection of a large number of representative installations reveals an inherent weakness in this respect. A plant might be "gold-plated" but its results could be limited by only two major factors:

First, the extent to which facilities have been provided for intelligent operation and effective, reasonable ease of maintenance.

Second, the ability and interest of the operating staff (including those who approve expenditures for maintenance) to conscientiously, unfailingly and economically produce the results for which the system was purchased.

Whether the plants are on the order of the Metro-

*Air Conditioning Engineering Specialist of the staff of Clyde R. Place, Consulting Engineer, New York City.

politan Life Insurance Company's new home office building with 1200 tons of refrigeration; the numerous plants in Rockefeller Center ranging from 300 tons to 1200 tons at a cost of nearly 3½ million dollars; or a 6 ton hotel cafe job in size, this writer's policy has centered on the very real consideration that these plants have to be operated and maintained rigorously and skilfully in order to fulfill their objective. Unless this is or can be done, the efforts of every party responsible for the system—from owner to designer—are not only apt to be wasted but productive of negative reactions on the part of the occupants.

While each plant differs in detail, the items listed below are common to most plants, and the practical value of this article lies in covering the main items to be checked or serviced in autumn, combined with an explanation of the reasoning behind these recommendations. At the same time it is hoped that certain considerations will prove valuable in suggesting improvements for existing plants.

Running Refrigeration Plant Intermittently

In running the refrigeration plant at more intermittent and brief intervals as the transition from summer to fall seasons takes place, it is a common mistake to assume somewhere along the end of August or first of September that the refrigeration plant has seen its last turnover for the season. Don't be fooled by a cold snap. Even in the northern United States we have sustained high outdoor wet bulbs as late as the last week in September.

You might be shut down for a week in the beginning of September, but the records show that during most any time in September a sustained outdoor wet bulb of over 72 F invariably occurs, when the refrigeration plant should be legitimately operated. But after being idle for a week, don't start up the machine without a thorough checking. To do so is dangerous business. You may argue that it is poor economy to run the refrigeration plant in September, and hold over until that time. Unless you intend to let go of a licensed engineer, you are probably wrong. To prove this contention, compute the total costs, including fixed costs, chargeable to one refrigeration ton-hour, and convince yourself that the use of the machine for an additional 40 hours or so gives a better investment balance in relation to the total hours the plant has previously been operated on the average in summer.

Keep the refrigeration in condition until the end of September so as not to get caught short, or produce an excessive relative humidity, productive of mugginess, which wet bulbs over 65 will do.

Preheater Bypass

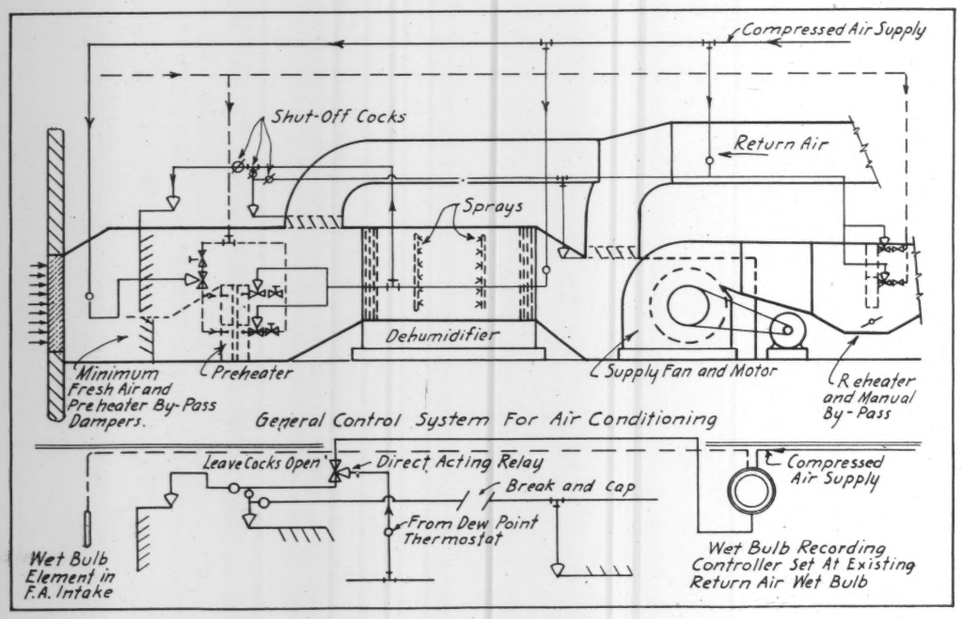
Most systems are equipped with an automatically modulating preheater bypass for permitting the full volume of air up to the dehumidifier capacity to be taken from outdoors. In this connection be certain of three things.

First, be sure that this bypass is closed off tight whenever refrigerating.

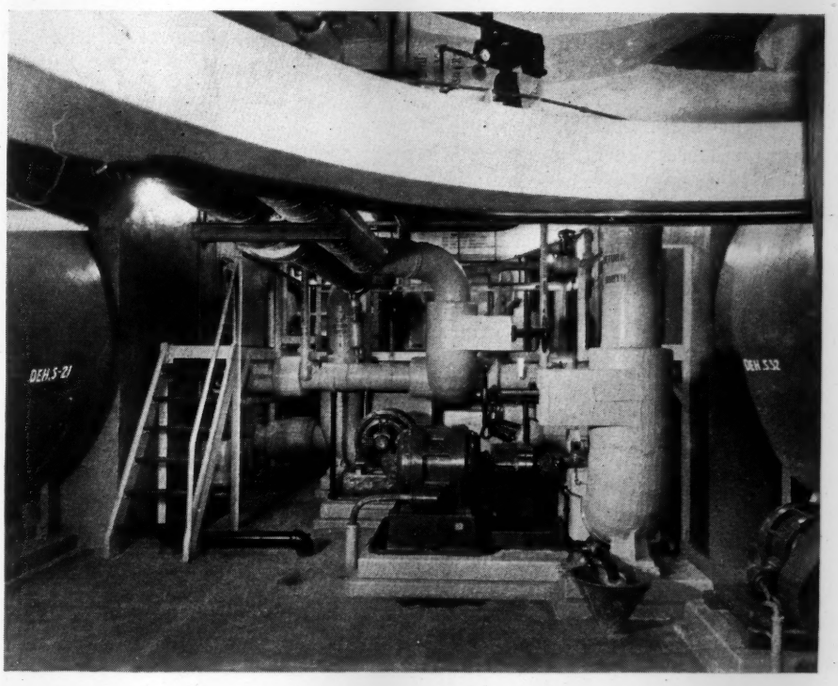
Second, be certain that you are getting the full rated volume of air through the dehumidifier when taking maximum outdoor air with the return air damper closed off or closed minimum, as the design and proper adjustment requires. This is important in realizing the maximum evaporative cooling capacity for best results and economy and bears investigation because of the relatively high dehumidifier resistance as against using a mixture of outdoor air and partial return air.

Third, just before transition from fall to winter takes place, that is to say, when the outdoor wet bulb goes down just below 32 F, check this bypass again, to be sure it is closed. Otherwise the combined preheater and return air capacities may not be sufficient to prevent a freeze-up.

There are two good methods which assure automatic functioning in accordance with the above three requirements. Either or both may be superimposed on an existing control system. Doing so, however, does not imply that periodic checking up of these three func-



Schematic diagram illustrating method of control for automatically securing maximum summer economy and reduced danger of winter freeze-up



Air conditioning plant designed by the author employing control indicated in diagram showing outdoor wet bulb control. The chilled water pumps in the foreground are completely overhauled each autumn

tions can or should be obviated. The first method involves the control of the preheater bypass damper so that it opens at all times when the outdoor air wet bulb is below the return air wet bulb—but doing this in a manner such that the dewpoint control—such as is conventionally provided—continues to control this preheater bypass in modulating positions, when the dewpoint is below dewpoint thermostat.

The second method is less expensive, but not so economical, and has no bearing on any of the other controls. (See diagram). An electric switch is tied in across the refrigeration motor starter—or some other element of the equipment such as a recirculation pump motor starter if the refrigeration is of the steam jet type. This switch is tied in either fully electrically, or with pneumatic controls—electro-mechanically—so that whenever the refrigeration is operated the preheater bypass damper is automatically forced closed—regardless of what the other controls call for. With this, checking is still warranted to see that this damper is closed with lower outdoor wet bulbs, such as occur in October.

On some systems some sort of automatic throw-over for summer to winter is provided. Theoretically, such a transition device is impossible while securing maximum economy. In any case, the intermediate (spring or fall) season provides opportunity to do some checking to get better economy, and to see that dampers and all matters pertaining to heating equipment are in proper positions and working order.

Shutting Down Refrigeration Plant

At the end of September the refrigeration equipment can be safely put up for winter. It is impossible to touch on everything for all types of equipment, but some of the major items with less thought of the details are in order.

One of the first steps, particularly with a new job, is to inspect the condenser and cooler tubes for slime, scale or corrosion. Steel tubes certainly need annual inspection. Dependent on water and other conditions, admiralty or bronze tubes, after repeated annual inspections, may not need cleaning or even inspecting more often than every three or four years. However, a conscientious engineer will want to know exactly what is taking place in these important parts of the equipment, and annual cleaning is bound to produce more efficiency and less expense. The latter is particularly true when using a cooling tower for condensing water.

A good idea is to extract a gallon of refrigerant and boil it off, preferably in a laboratory, to check the oil content. This should be done annually. One cannot know by merely looking, but a little oil does a great deal to cut down heat transmission. ,

By hand, or other means, turn the compressor over once a week, through at least one full revolution, allowing it to rest in a different position each week. Mark some part and number these marks with chalk to be certain. Do this throughout idle periods.

An outline of general procedure is as follows:

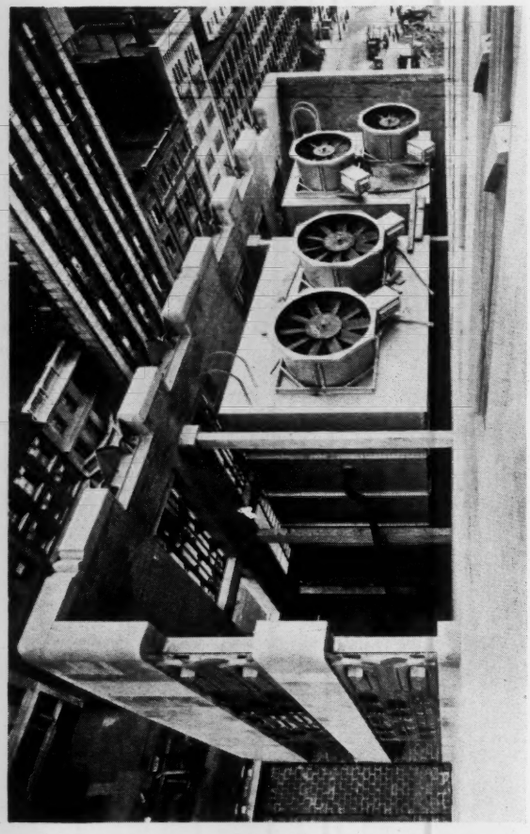
First, remove all refrigerant and drain out *all* water. A small hand crane or hook eyes in the ceiling facilitate the yearly removal of heavy parts. When such measures are taken by the chief engineer or management, most of the inertia to proper servicing is overcome for future work which should be considered just as necessary as starting and stopping equipment.

In reciprocating compressors, remove excessive vibration in the machine and high pressure lines, especially at joints. Incidentally, make several full load computations on the basis of Btu per minute and figure the ratio of effective thermal work by cooler plus refrigeration power input to the Btu per minute condenser water output.

Multiply the motor horsepower by 42.5 and apply the most intelligent factor possible to get the portion of motor power absorbed by the condenser water. This

factor will vary between 0.5 and 0.65 depending on the motor efficiency under the loading and power conditions, and heat loss by the equipment to water jacket and the surrounding air.

The main considerations immediately after final shutdown with reciprocating compressors of any type, are checking the end clearances and the seating conditions of the suction and discharge valves. Failure to perform



Cooling towers require attention in autumn for winter protection

these duties by competent refrigeration mechanics is wasting money. Refrigeration is the largest single element of expense. Therefore, a proportionate amount of maintenance to keep efficiency as high as possible is essential.

A wide latitude in efficiency exists between a machine in perfect condition and one on the verge of breakdown. It is usually the aggregate of small items on refrigeration which reflects in appreciable power and water waste. Regardless of type or apparent simplicity of the oiling system, take these down entirely, blow out all lines, remove stuffing box and repack. To use cheap and possibly inferior oils on a refrigeration machine is not economy.

Regardless of type of compressor using a chemical gas refrigerant, in the best practice in air conditioning maintenance, tubes are tested individually by compressed air. A great deal of extra work? Certainly it is. But in plants where shutdown becomes a calamity and plants which have a keen sense of pride in their equipment, such annual pressure tests of tubes are a part of the refrigeration shutdown routine. A special fiber brush should be obtained for tube cleaning to fit size of tubes.

On centrifugal compressors the following additional matters should receive attention:

All bearings must be opened and adjusted if necessary. The evacuator and liquor pump must be opened, unassembled, cleaned out thoroughly and replaced. The rotor end seals should be examined. Each plant to be properly serviced must consider these items.

All condenser and chilled water pumps used only in conjunction with refrigeration should be opened and motors blown out with compressed air.

The Cooling Tower

If a cooling tower is employed, the lines need complete draining, the fan bearings examined, filling examined, nozzles removed for cleaning and any interior iron work such as filler suspension clips wire brushed and repainted. The fan impellers, if metal, are to be scraped and repainted; water basin checked for corrosion *under* the asphaltum or bitumastic and recoated if necessary. Motors should receive very careful inspection, even the totally enclosed type. Protect the vital equipment by boarding up with removable, tight, sheet metal housings. Be certain that make-up water lines are thoroughly drained or covered with adequate insulation. In this connection watch for early cold snaps over night.

In place of ordinary grease, a coat of petroleum jelly brushed on all unpainted metal parts of refrigeration, is recommended to maintain a better appearance.

Checking the Heating Equipment

Attention should next be given to the steam equipment. Depending upon the design, it is possible that steam has been used right along for reheating. But now is the time to check all this equipment—that is to say, before it might be overlooked. Therefore check the condition of the rubber diaphragms on all compressed air damper motors and steam operated valves, if compressed air control is used with valves having rubber tops. If metal tops are in use check the ability of the valves and dampers to modulate and note the pressures and range between full open and full closed, checking these pressures against any other equipment operated in sequence.

Every trap in the system should be opened for inspection and cleaned out. This is important for many reasons. Difficulties of a subtle nature sometimes occur. For example, a leaky trap has been known to heat up a branch return line, into which another return line connected, and as a result when the steam from the second-mentioned heater was supposed to be turned off, a rise in air temperature of several degrees due to conducted heat from its hot return, was caused by this heater. This was extraordinary, of course, but serves to disclose possibilities resulting from a trap not closed tightly. Wasted steam, particularly with vacuum return, is also a result of a dirty or poorly seated trap.

Usually two rows of preheater units are provided. At the first opportunity, it is essential that the second row of preheater, usually under thermostatic control, operates to hold water temperature, only after full volume of return air is taken, as a matter of economy.

Do not neglect turning on of outside row by hand when the first cold snap below 36 F occurs.

Be sure that the air volume is where it should be, so as to obtain the maximum residual room pressure possible to offset infiltration. Incidentally do everything reasonable to minimize infiltration. This expensive phenomenon produces localized conditions such as a draft. Do not expect to kill all infiltration, at all times, in outside rooms, rooms adjacent to corridors or any other room in a building over six floors, by means of residual pressure from the fan system over and above the mean barometric pressure around exterior of building. However, blocking return air down somewhat for the winter season is helpful.

In rooms heated by radiators, air should be delivered by the air conditioning system at a constant temperature equal to room dry bulb temperature, providing, of course, the radiators are under thermostatic control.

When heating on the first cold day of 30 F or below, check the temperature by testing at the point where the supply duct has no further apparatus interposed in it and again at the room supply outlet of this duct or branch ducts. Select uncovered ducts for this test, and by noting the temperature drop and measuring the cfm, figure the heat loss and determine whether insulation would pay. In any case, compensation must be made during operation if the control is of a constant temperature supply type. Investigate the possibilities for such losses in the name of economy.

Better heat transmission efficiency is obtained when the air volume is higher. Therefore set any manually adjusted heater bypasses in the best position without imposing excessive reduction of the total volume of air delivered.

Check dew-point control making certain it is functioning within suitable narrow temperature limits of the order of plus or minus one degree. A not too slow action is also important. Control as close to 32 F as deemed safe in instances when desiring to keep condensation to a minimum on the inside of windows.

For further reduction of dew point below a safe temperature of, say 37 F, try valving off one bank of sprays. If this is not successful, run a flooding line from the pump with perforations across the top of eliminators and cut off sprays. This provides some humidification and cleaning efficiency with wetted eliminators almost as good as full sprays. Of course, operating in this manner, delivered air should be filtered for best cleaning results as winter air contains more soot.

All matters discussed herein may not exactly apply to every plant; thus it is impossible to be all-inclusive. Nevertheless, all those concerned with proper continuous performance of air conditioning plants will appreciate the considerable amount of work which should be scheduled in log form as part of the annual maintenance routine. Genuine effort, adequate budgeting and pride in careful supervision every autumn is necessary to obtain best results.

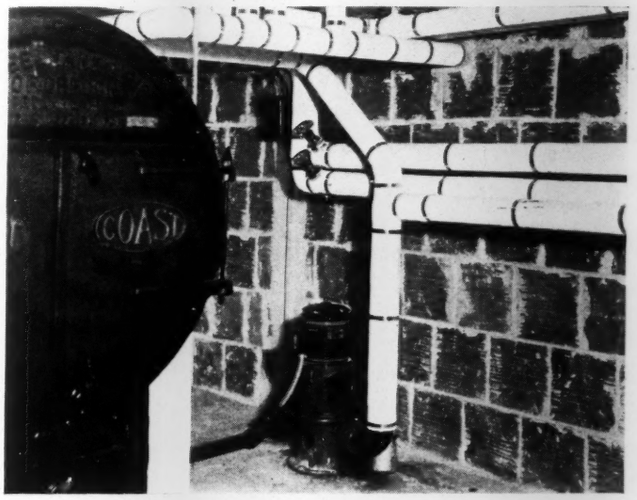
Modernize Means *Profit*

to Many Apartment Buildings

By **H. R. Coleman***

MODERNIZATION of heating plants in income-producing properties such as apartment buildings represents a self-liquidating investment as it puts such properties in better position to compete for tenants and maintain rental scales, in addition to reducing operating costs in a great many cases. The soundness of expenditures for this kind of modernization work is recognized by trustees and insurance companies, which have come into the ownership of apartment buildings in every part of the country. That organizations of this kind have accepted the idea of modernizing should be carefully considered by building owners and engineers, because if an investment can put a building on a paying basis and the necessary money is

*Stevens-Root Company, Chicago.

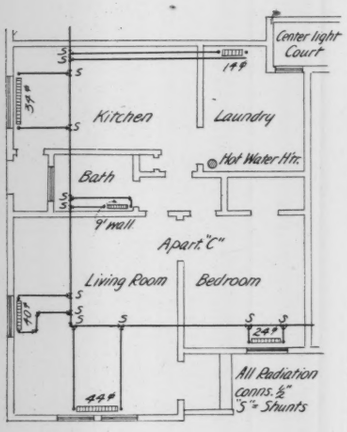


The circulating pump and boiler for one of the groups

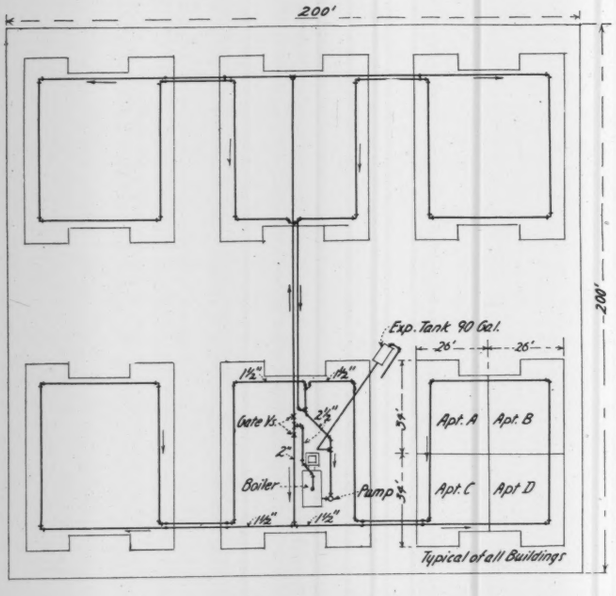
or can be made available, it is wasteful to overlook the opportunity.

The modernization work itself naturally varies with each building, depending upon the requirements of each case as determined by a survey. Articles in HEATING, PIPING AND AIR CONDITIONING during the past two years have given information on a wide variety of specific cases. In the May, 1934, issue (for instance) the modernization of a five-building apartment group near Philadelphia with a small central steam plant was described. Among the advantages that were obtained were lowered operating expenses and 95 per cent occupancy instead of 60 per cent. Before modernization taxes were unpaid and mortgage holders were getting no return; after, taxes were paid and interest payments were made in full.

The job described in the present article is likewise an apartment group, smaller than the project described in May, located in Portland, Oregon. Not only is the work done and the results achieved of interest in themselves, but this example serves to show the wide variety of buildings in which modernization work is justified. The work was done when an insurance company became the owner of the group.



Piping and radiation in typical apartment. Shunt fittings, installed at each radiator connection, direct the proper amount of water to each unit of radiation



Lay-out of the piping for one of the groups of one-story, four-apartment buildings

The method of heating the Granada Court Apartments, Portland, is novel in some respects, particularly because the 44 individual 3-room apartments, instead of being in one building, are in eleven one-story structures. Six 4-apartment buildings comprise one block, and five 4-apartment buildings the other. Heating is by means of two closed, one-pipe, forced-circulation hot-water systems, one for each group and each alike. The six-building block has 144 four-tube, 26-in. radiators with a total of 3736 sq ft of radiation, and the five-building block has 120 radiators with 3100 sq ft of radiation. The radiators have packless valves. Maximum temperature of the water circulated is 240 F. Each building was formerly heated by a gas-fired furnace.

The oil-fired steel boiler for the larger group is rated at 9840 sq ft, and that for the smaller group at 8320 sq ft. These are based on the *A. S. M. E.* code for water rating, whereas in using a closed hot-water system the load is rated the same as for a steam system. A 1½-hp centrifugal pump circulating 90 gpm is used with each system. Temperature control for each system is by means of a thermostat to start and stop the pump. Each thermostat is located in the north-east corner of the living room of the building in which the boiler is located.

The hot water mains are installed in 200 ft of concrete tunnel and are insulated with 4-ply asbestocel. One of the diagrams reproduced here shows the arrangement of the piping from one of the groups, and the other shows the piping and radiation in a typical 3-room apartment. The shunt fitttings shown in the drawings are installed to positively direct the required amount of water into and out of each unit of radiation and to make possible the use of a one-pipe system.

Domestic hot water (circulated by two ¼-hp pumps) is supplied each group by a copper U-bend heater, water from the boilers being circulated through the coils while circulation to the radiation is prevented by a trap in the main except when the thermostat calls for heat. The heater for the No. 1 group has 30 sq ft of surface, and that for No. 2 group, 35 sq ft. Water for domestic use is supplied at an average temperature of 160 F.

Fuel oil to heat these two groups of buildings and to supply them with service water cost about $75.00 per month during the last heating season. The winter was mild, with an average temperature of 47 F, the lowest being 34 F. An inside temperature of 73 F was maintained. In addition to savings in operating and maintenance cost, this modernization work improved comfort and consequently rentals.

Check Compressed Air Piping

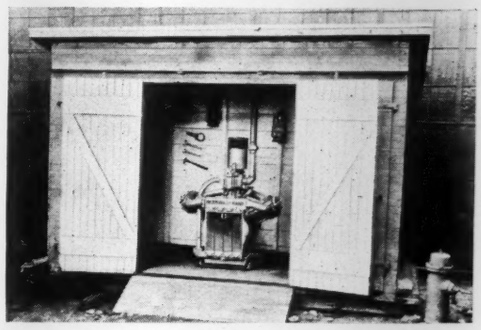
to Stop Waste, Excessive Cost

Cost of Air, Steam and Water Leaks

Size of Opening	Air: No. of cu ft wasted per month 100 lb pressure	Air: Total cost of waste per month at 10c per 1000 cu ft	Steam: No. of lb wasted per month 100 lb pressure	Steam: Total cost of waste per month at 60c per 1000 lb	Water: No. of gal wasted per month 40 lb pressure	Water: Total cost of waste per month at 1.5½c per 1000 gal
3/8"	9,979,200	$997.92	460,000	$276.00	692,400	$108.00
1/4"	4,449,600	444.96	203,000	121.80	307,700	48.00
1/8"	1,114,560	111.46	50,500	30.30	76,900	12.00
1/16"	278,640	27.86	12,750	7.65	19,200	3.00
1/32"	69,552	6.96	3,175	1.91	4,800	0.80

By **A. W. Loomis***

Small air-cooled compressor in a tool shed supplying air beyond the regular shop lines



***INCORRECT LOCATION** of compressor intake, no aftercoolers or receivers, piping too small, leaks, water in the lines . . . all result in high compressed air costs. Compressed air costs real money; a check-up of the equipment and the piping will show opportunities for saving*

A CHECK-UP of compressed air piping is a paying proposition in many plants, few of which have air piping systems that function perfectly. Usually such systems have been added to and revised many times since the original installation was made. In "gradual growth" of this kind many parts of the piping lay-out are likely to be inadequate to serve present needs. As has often been pointed out in these pages, this is a situation met with in most types of industrial piping installations.

One plant engineer has stated that proper attention to the piping will in a great many instances reduce compressed air costs from 5 to 25 per cent. Let us see what savings can be made by correcting typical faults.

Location of Intake Piping

The compressor intake may be in a hot engine room, and where this is the case the air is cooled and therefore contracts before it reaches the point of use. Thus, there is actually a smaller volume of air available for use than was taken into the compressor. A saving in power can be made by placing the intake in the coolest possible place.

For example, a compressor delivering at 100-lb pressure 3,000 cfm of free air[1] operates 8 hours a day, 300 days a year, at an average load factor of 70 per cent. Its intake is in an engine room where the average temperature is 85 F. The temperature in the shops where air is used averages 70 F. At a cost of $50 the intake can be extended 5 ft through the roof where it will obtain air at an average temperature of 60 F.

With the intake in the engine room 1,028 cu ft of air is taken in for each 1,000 cu ft delivered at 70 F. When the intake is moved outside only 981 cu ft need be taken in to deliver 1,000 cu ft at 70 F. This is a saving of 4.6 per cent.

Assuming that all other charges are fixed and only the saving in power costs is realized, this will amount to about $1030 in a year's time with electricity at 3 cents per kw-hr.

Aftercoolers and Receivers

Aftercoolers and receivers are an important part of the air-piping system. The purpose of the aftercooler is to remove oil and water from the air. The receiver acts as a reservoir to absorb and eliminate pulsations in the air flow; in addition, it aids in removing oil and water. Any experienced engineer knows that considerable water accumulates in compressed air lines. However, it is not always realized just how much water there actually is or how great is the damage which it does.

Under saturated air conditions such as obtain in foggy and rainy weather the condensation from each 1,000 cu ft of air at 70 F delivered at 100 lb per sq in. by a

*Ingersoll-Rand Company, Phillipsburg, N. J.

[1]*Free air* is air at the temperature and pressure conditions existing at the compressor intake. It is the usual method of expressing compressed air quantities.

single-stage compressor amounts to 1.0 lb, or in the case of a 1,000-cfm compressor 7.2 gallons per hour. The intercooler of a two-stage compressor will remove part of this moisture, but even then 1.9 gph will be removed by the aftercooler.

This moisture will be carried into the air lines unless an aftercooler is being used. Once in the air lines it gets into tools and washes out the lubricating oil. It also causes air wastage, for workmen will invariably open an outlet valve wide to blow out the water before connecting the tools. Pipe lines are corroded by the condensate, not only reducing the life of the lines but also injuring many tools because particles of rust are carried into them from the main lines.

The air receiver should be of ample size to dampen pulsations effectively and to aid in removal of moisture. A safe rule to follow is to make the receiver equal to or greater in volume than the volume of compressed air delivered per minute. For example, a compressor (which operates at 100 lb per sq in. discharge pressure) has a piston displacement of 450 cfm. According to the above rule, a receiver of $450 \times \frac{14.7}{100 + 14.7} = 58$ cu ft capacity should be installed. It is safe to say that the annual savings effected by an aftercooler and the use of a receiver of ample size as against one of inadequate size will amount to several times the first cost.

Check the Piping System

A complete check-up and modernization of the compressed-air distribution system involves several factors. It is necessary first of all to find out whether the piping is of ample size; the importance of this is not always realized. Water piping, if too small, increases pumping costs and this effect is easily measured. Ammonia piping of inadequate size increases refrigeration costs and this, too, is fairly obvious. On the other hand, the loss caused by under-sized air transmission lines, while just as great, is not so easily determined. Under-sized lines will cause a considerable drop in pressure between the air receiver and the point of use.

For instance, the air may leave the receiver at a pressure of 100 lb and reach the point of use at 80 lb. Most air-operated tools are designed for use with 90- to 100-lb air. Reduction of the pressure to 80 lb will result in loss of efficiency and slowing down of the tools.

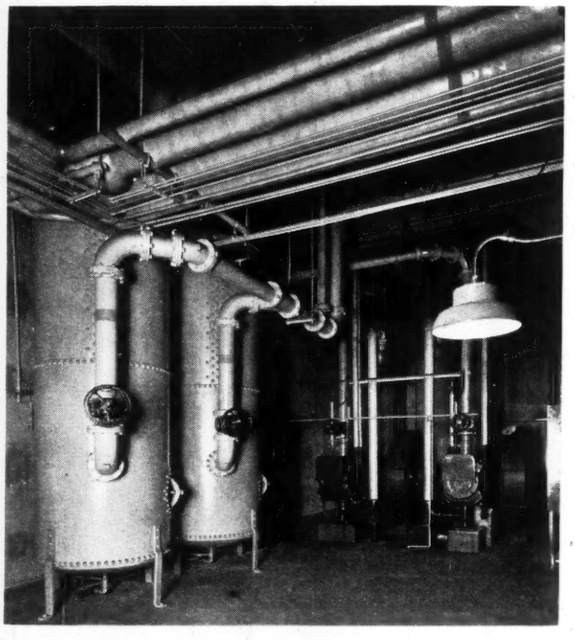
While it does not hold in all cases, it is in general true that a reduction of pressure from 100 to 80 lb per sq in. will reduce the available horsepower in a pneumatic tool some 20 per cent. At the same time it reduces the speed from 5 to 25 per cent. This means that it takes considerably longer to complete a job when the pressure is low.

Still another method of looking at the matter is to consider the excess power required to overcome the friction of small pipes. This allows close computation and will give some idea of the costs involved.

For example, in a certain plant the pressure at the point of use is 95 lb per sq in. while the compressor is delivering air at 125 lb. A reasonable pressure drop for this system would be 5 lb. In other words, with adequate pipe sizes the compressor could operate at 100 lb discharge pressure and 95 lb pressure would be available at the point of use.

With a two-stage compressor it requires about 213 hp to compress and deliver 1,000 cu ft of free air at 125 lb pressure. At 100 lb pressure about 190 hp is required—a decrease of 10.8 per cent. A reduction of 10.8 per cent in the power bill amounts to a considerable sum. A compressor delivering 2,000 cfm of free air at 125 lb discharge pressure and operating 8 hours a day, 300 days a year, at 80 per cent load factor will, at 3 cents a kw-hr, require about $18,300 worth of power per year. A saving of 10.8 per cent amounts to $1975.

In an existing distribution system the pressure drop may be measured by installing a pressure gage near the point of use and noting the difference between it and the gage on the air receiver. This reading must be taken *while air is being used;* otherwise pressure will build up until it is equal in all parts of the system and no pressure drop will be noted. By this method of measurement, actual figures can be obtained for normal operation, peak loads, or any desired condition.



Compressor and air receiver installation in a large office building

If the friction drop is excessive, existing pipe lines should be replaced, or made adequate by the addition of parallel lines. It seems hardly necessary to add the caution that two 3-in. lines do not equal a 6-in. line. The cross-sectional area of a pipe (and therefore carrying capacity) varies with the square of the diameter; hence it takes four 3-in. lines to equal a 6-in. line.

When two or more lines serve the same point, the flow in the lines will be such that the pressure drop is equal in each line.

If the quantity of air flowing through a given pipe line is unknown, it may be measured by one of several types of air meters. All such devices are subject to errors, which under some conditions may amount to 6 or 7

per cent. However, they are accurate enough for most calculations of the nature being considered here.

Leaks are one of the most common faults of air piping systems. They are allowed to exist because they seem insignificant or because their actual cost is not realized. If the actual loss in dollars caused by leaks were known to each plant manager, there would be much more attention paid to the proper care of pipe lines.

It has been observed that a great many plants have leakage losses equal to as high as 10 per cent of their total compressor capacity. In a plant using a compressor having a capacity of 1,500 cfm of free air, the annual usage probably amounts to 150,000,000 cu ft or more, and leaks would account for 15,000,000 cu ft. If 80 per cent or 12,000,000 of the leakage could be eliminated, there would be a saving, based on a cost of 10 cents per 1,000 cu ft, of some $1,200. Some plants have a compressed air cost of less than 10 cents per 1,000 cu ft; many have a greater cost. At 3 cents per kw-hr the power cost alone for compressing air to 100 lb per sq in. discharge pressure will amount to around 7½ cents per 1,000 cu ft of free air compressed and delivered.

The actual amount of air wasted by leaks can easily be determined by noting the compressor capacity required to keep the air pressure constant during the night or on a holiday, when no air is being used.

It should not be a difficult job to locate and correct most air leaks. Piping that is properly supported and put together should not leak at screwed connections. Flanged connections often develop leaks because of deterioration of the gaskets. Such joints should be regularly inspected and repaired. One of the most frequent means of loss is through leaky valves. There are usually many of these in an air piping system. All faulty ones should be repaired or replaced. Cheap and inefficient valves are false economy.

No leak should be neglected because it appears too small. At a cost of 10 cents per 1,000 cu ft, a hole of $\frac{1}{16}$-in. diameter will waste $28 worth of 100 lb air per month. Compressed-air is much more expensive than either steam or water.

In many plants air lines are now being welded to eliminate as many joints as is possible.

Other Possible Savings

Where water collects in the pipe lines much air is wasted in blowing out the water before tools are connected. Even where ample aftercooler and receiver capacity is installed, provision should be made to drain the line at various points toward which any moisture will naturally drain. Service taps from the main line should be made with the tee pointing upward. This will prevent moisture in the main line draining into the service line.

Drills, riveters, and other air-operated tools are often left after working hours without closing the valve on the service line. This does no particular harm if the valve on the tool is tight. In many cases these valves pass considerable air, since they are designed to control the tool rather than to hold the main-line air pressure. It does not take many such leaking tools to waste a considerable quantity of air.

Stopping this practice may require considerable education, but there is no doubt that it is worthwhile. In one shop considerable success in reducing leaks of all types was attained by posting in conspicuous places tables showing their cost.

A check-up of a plant's air piping is likely to uncover a need for air in an isolated building to which it would be costly to extend the piping system. Or, it may be found that a large compressor must be kept running uneconomically at part load to operate control instruments or supply some department that works at odd hours. In other cases, a small amount of air may be required at a pressure different from that supplied by the regular system.

In such cases it may be economical to use one or more small compressors in addition to the larger unit that furnishes the main air supply. Such units are available in a variety to suit practically any need.

Carbon Dioxide Gas in Molasses Complicates Piping and Pumping

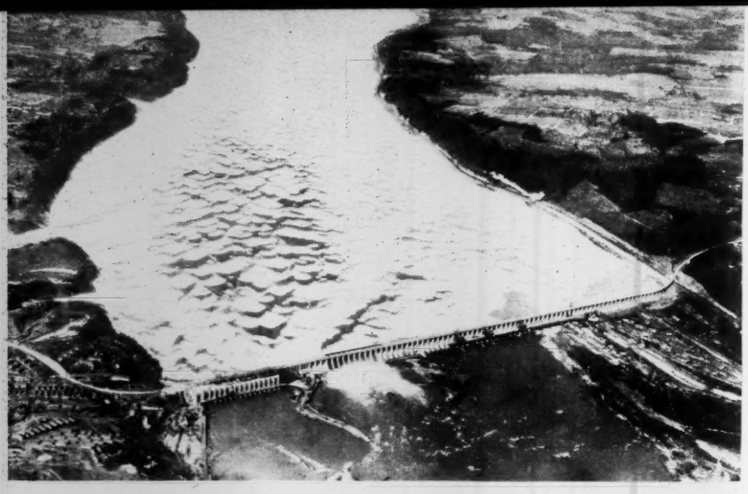
WE READ with interest the article[1] on pumping molasses in the May issue of H. P. & A. C. So far as we know this is the first time this matter has ever been discussed in print with such detail and undoubtedly many plant engineers appreciate having the information available. So far as a casual reading is concerned the information is correct and quite comprehensive.

If a further discussion on this subject is to be taken up, we would suggest a discussion of the peculiar characteristics of molasses on the suction side of pumps and the necessary arrangements that have to be made to take care of this problem. Not only does all of the thought have to be given to the problem that has to be given to the pumping of any viscous fluid (such as tar, asphalt, or heavy lubricating oil) to insure pipe lines which will allow a sufficient quantity of material to reach the pump, but in handling molasses consideration has to be given to the fact that black strap has a very considerable quantity of CO_2 gas dissolved or entrained therein. One plant made a laboratory test and found that 1 cu ft of molasses under a 15 inch vacuum would expand to 1¼ cu ft and apparently be just as solid a mass as it was when not under a vacuum. This was caused by the expansion of the gas bubbles inside of the molasses.

Obviously if this is the case, then high vacuum on the suction side of pumps is something to be avoided in handling black strap molasses. Unless the vacuum and velocity are kept extremely low, gas will be extracted from the molasses and travel through the pipe lines at a much higher rate of velocity than will the sluggishly moving molasses. The resulting water hammer loosens pipe joints, and, to say the least, is annoying to everyone within hearing distance.—Wm. E. Worcester.*

*Vice-President. Kinney Manufacturing Company, Boston, Mass.

[1] "Piping and Pumping Molasses." by Lee P. Hynes, Heating, Piping and Air Conditioning, May, 1934, pp. 197-200.



Airplane view of Conowingo Dam showing masses of ice which accumulate in winter. Power house is at extreme left side of the river. Further to the right may be seen ten of the regulating gates partially opened. The excess flood water can be seen rushing through these gates, which are kept free for operation by electric heaters located in wells in the masonry

Electrical Power Industry *has many*

By **Lee P. Hynes***

***SO GREAT AND VARIED** an industry as the generation and distribution of electrical power has many special problems in heating and air conditioning. The methods used in solving these problems are equally useful wherever low temperatures, ice,*

PRESIDENT ROOSEVELT'S recent inspection of the great hydro-electric power projects on the Columbia River and his radio address from Glacier National Park have focused public interest on the development and control of the electric power industry. The government is now conducting a large scale experiment in power generation, distribution and sale in the Tennessee Valley and will soon undertake a somewhat similar project at Passamaquady, Maine, the purpose being to secure further data to guide in formulating plans for developing and controlling the electrical industry. The government is also building or financing several great hydro-electric and flood control projects among which are Boulder Dam on the Colorado River, the Central Valley project in California, the Bonneville Dam in Oregon, the Grand Coulee Dam in Washington, and the Fort Peck Dam in Montana. It is also actively negotiating plans for large power developments on the St. Lawrence River and in numerous other localities.

The future of the power industry and the part it will play in the ultimate solution of our economic and social problems is the subject of so much thought these days that perhaps it will be of interest to present some general facts before we take up our discussion of heating and air conditioning problems in this industry.

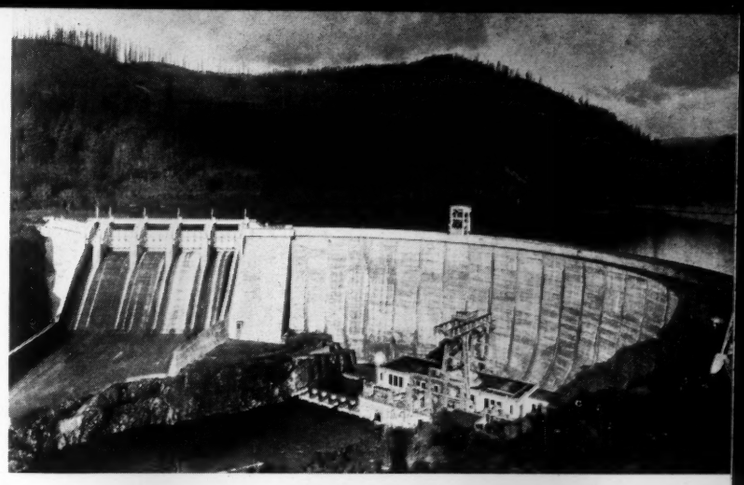
Electric power is energy converted from other forms, principally by means of steam boilers, diesel engines and hydraulic turbines. Highly developed industrial countries are making rapid strides in the use of their available water power. The United States leads the world with about 16 000,000 hp in use. Canada and Italy have 7,000,000 hp each; Japan, France, Germany, Switzerland, Norway, Sweden, Spain, and Russia follow in order. Russia has the largest single hydro-electric power plant in the world, of 810,000 hp capacity on the Dnieper River. The largest plant now operating in the United States is at Niagara Falls, with 452,000 hp and there is also a plant on the Canadian side of 560,000 hp making the total power developed by the United States and Canada at Niagara 1,012 000 hp. Boulder Dam when finished will be rated at 515,000 hp installed with 1,835,000 total hp available. The greatest potential power development is the Grand Coulee Dam on the Columbia River in Washington where 2,660,000 hp will be available, although the present project will develop only 147,000 hp. It seems certain that hydro-electric power will become an increasingly important factor in the electric industry.

There are two extreme types of hydro-electric plants. One is the high head type such as Boulder Dam, where the maximum water level will be 584 ft above the turbines. Water will be brought down to the turbines through four great steel penstocks 30 ft in diameter. The other is the low head type such as Conowingo, Md., or Safe Harbor, Pa., where water is admitted directly to the turbines through special intake tunnels in the dam itself.

When dams are built across rivers it is very important to control the water level above the dam in times of flood by opening regulating gates so that surplus water may escape to the river below; otherwise it would rise high enough either to sweep ruthlessly over the dam or else flood the surrounding country. In any case, the power plant must be located on the downstream side of the dam, so as to receive the full head of water pressure at the turbines, and it must be protected from flood damage. Electric current from generating stations is delivered to transformers and distributed by high voltage transmission lines leading away to distant communities. At convenient points there are local substations which serve to reduce and regulate the voltage and distribute the power to industrial and domestic users.

Heat and humidity problems are met in every stage of the power industry. Ice is a menace when it collects above the dams, freezes up the gates or blocks the

*Consulting Engineer, Philadelphia, Pa. Member of Board of Consulting and Contributing Editors.



Ariel Dam of the Inland Power and Light Company, Portland, Oregon, from the downstream side. At the left are regulating gates for discharging surplus flood waters. The power house is shown at the base of the dam, water being admitted to the turbines through intake tunnels through the dam itself. The steel tower at the extreme right supports the transmission cables. This entire power house is heated by fan circulation type electric unit heaters

Heating and Air Conditioning Problems

Methods Used Suggest Applications in Other Fields

excessive condensation, etc., must be dealt with. Plant and building engineers who feel they must keep alert to progress in fields other than their own will welcome this article as an interesting explanation containing many useful suggestions.

intake racks which protect the turbines. Sleet overloads transmission lines until they break and interrupt service. Condensation forms on the walls of control rooms, substations, and cable vaults, and on switchboards and instruments causing stray currents to travel where they are not wanted. Oil filled cables, transformers, and disconnecting switches are endangered if their insulating oils absorb moisture. Motors for pumps in drainage wells and pumping stations must be kept dry and if floods or accidents wet electrical equipment, thorough drying out is necessary before service can be resumed.

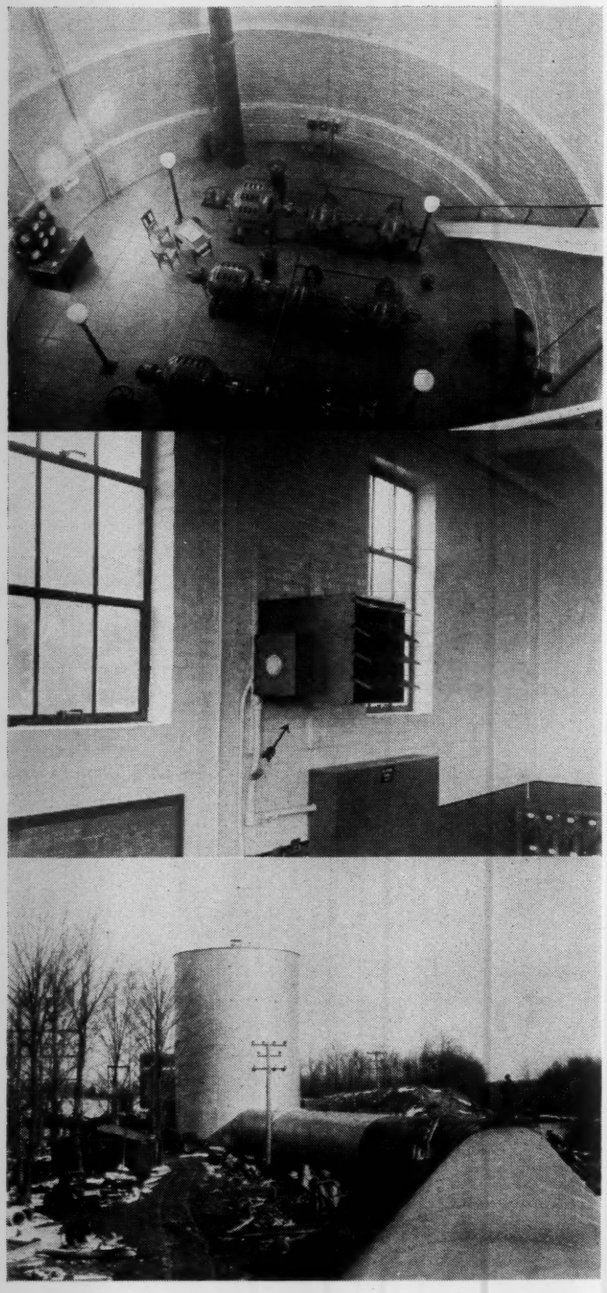
How Ice Is Combatted

A low head dam across an active river is a massive structure of reinforced concrete with many large gateways through it. In these gateways are fitted steel regulating gates, frequently as wide as 40 ft and built in the form of a rectangular box 3 to 4 ft thick. These gates are reinforced with structural steel members and have roller bearings at each end which engage heavy steel rails, set vertically in recesses, in the masonry of the dam. The gates thus interlock with the dam itself and are able to withstand the tremendous water pressure against them. They are raised and lowered by means of powerful motor driven hoists and the openings are kept adjusted properly to discharge all excess flow of water. Gates of unusual height are constructed in sections consisting of upper and lower leaves so that each may be operated independently, because the mechanical load of handling the entire gate in one section would be too great.

The vertical water seal between a movable gate and a stationary dam is made by means of a flexible bronze strip on the gate which the water pressure forces tight against a seal plate on the dam. A similar seal is used horizontally between the leaves of sectional gates. As much of the downstream side of a gate is exposed to the air, any leakage through the seals will freeze in winter and snow and sleet also adhere until the gates become frozen so solidly in their grooves that they can not be operated. Even if sufficient power could be applied to pull a gate loose from the ice, the seal strips would be torn loose and destroyed.

It is a difficult heating problem to keep these seals and roller bearings warm enough to prevent their freezing up, or to thaw out frozen gates before attempting to move them. A practical method has been found by providing vertical wells in the masonary back of the roller guide plates. This is usually done by imbedding 4 in. pipe in the concrete. For large gates these vertical wells may be as much as 80 ft deep, reaching down from the top of the masonry to a point below the bottom of the gate. Electric heaters enclosed in sectional waterproof casings are lowered down into these wells. At the top of the casing is a fitting for conduit connection so that electric cables can be connected to the terminals of the heater. A water tight plug is then inserted to close the top of the heater casing. The top fitting rests on the masonry and the heater casing hangs free within the well. Whenever weather conditions require, the heaters can be put into service by simply turning on switches. Heat is conducted through the guide plates to the roller bearings and through the masonry to the seals so that all the movable parts are kept free from ice.

In very cold climates such as in Canada, it is often necessary also to apply heat to the upstream face of gates, otherwise masses of ice will be frozen on the steel plates. Unless this is prevented, the gate may become so overloaded with ice that it can not be operated or if it can be opened, these masses of ice may loosen with the first mild weather and be wedged under the partially opened gate by the force of the river's flow. Gates have been badly wrecked and lives lost in this way. To overcome this dangerous ice formation, water-



TOP TO BOTTOM

Looking down into the pump pit at the Peoria Pumping Station, showing pumps driven by one 350 hp, one 500 hp, and one 700 hp electric motor. Locations such as this require moisture prevention

Close-up of a fan circulation electric heater for heating an oil purification room, or a station building. Ariel, Osage and other large hydro-electric power plants are heated entirely by electric unit heaters

For small plants in mountainous country, water is often carried a long distance through large steel pipes. When shutting down the flow to turbines, it is necessary to have large vertical surge tanks to absorb the shock of checking the water flow. These tanks must be protected from freezing over solid as their water level must be free to rise and fall rapidly. The pipe lines themselves must be vented at certain points, otherwise when they are emptied rapidly, sufficient vacuum would be caused to collapse them. These vents are kept free of ice by electric heaters

proof heaters are attached to the interior surfaces of a gate, at the points which need special protection. This depends upon the downstream water level and the exposure of the gate surfaces.

Wind driven spray will sometimes form ice masses in unexpected places and it is convenient to have some portable means for cutting this ice away before it does any damage. At Conowingo Dam, a unique equipment is installed for this purpose. It consists of a 1500 gallon insulated hot water storage boiler located in one of the portable gantry cranes which travel back and forth on the dam to service the gates. A feed pump on the footboard of the crane fills the tank, a suction hose being dropped into the water above the dam. The water is heated electrically over a period of several hours to distribute the electrical load, power being taken from the trolley wires of the traveling crane. This high temperature hot water is available for use when and where required and is discharged under high pressure through a hose and nozzle so as to cut ice formations loose from their attachments.

The intake tunnels which feed the turbines are well protected against floating ice by heavy steel rack bars set edgewise, close together. Floating ice and trash can be cleared away from these without great trouble. At times, however, the water itself is just at the freezing point and when it strikes the bars, the impact causes ice particles to form on their surfaces and build up until the intake is seriously impeded or completely blocked. To combat this, a low voltage current can be passed through the bars in series and as the ice is a fairly good heat insulator, the bar becomes warm enough for the ice to loosen from its surface and be swept away by the stream. Transmission lines suffer when sleet forms upon them in a similar manner. By sending a low voltage, high amperage current, through sections of the line which are endangered, the sleet can be loosened so as to fall off before the lines break under the weight of their ice load.

Overcoming Condensation Troubles

Condensation forms on walls, switchboards, machines and other objects under certain atmospheric conditions exactly as dew forms out-of-doors. Wherever humid air comes in contact with surfaces cold enough to chill it below its dew-point temperature, condensation takes place and water is deposited in the form of dew. Moisture is of course very harmful to electrical equipment because it may cause short circuits and other serious difficulties on the switchboards and in electrical machines. Excessive moisture is also damaging to the walls and is often objectionable in buildings other than electrical stations. Dew is not formed under all conditions, but only when there exists a definite relation between the relative humidity of the air and the temperature.

It would be possible completely to condition the air in any building so as to avoid condensation troubles but it would be unnecessarily expensive to provide and operate equipment for adequately dehumidifying the air in electrical stations, pump houses and so forth. The desired result can be obtained at less cost simply by applying sufficient heat to reduce the relative humidity and

raise the dew-point temperature enough to prevent condensation. For locations where operators are present all the time, it is of course necessary to provide sufficient heating to maintain a comfortable temperature of 68 or 70 F, and at these temperatures trouble is seldom experienced with dew formation. However, there are many isolated, unattended buildings such as automatic substations for electric power distribution or for telephone service, and there are also water works, sewage pumping stations, meter rooms, cable vaults, tunnels and so forth, where little, if any, heating is required except to prevent freezing and to avoid condensation troubles.

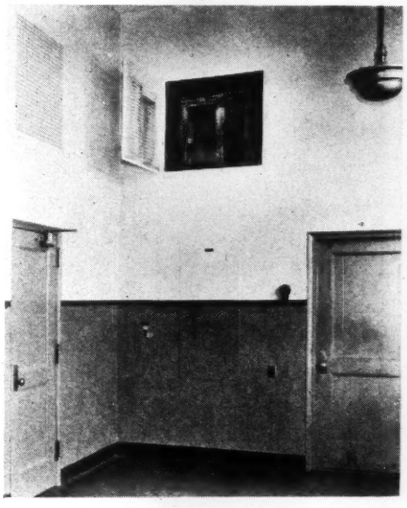
As the danger of condensation varies greatly according to weather conditions, the temperature required for its prevention is not a constant one. Frequently in cold weather an inside temperature as low as 35 F is ample to prevent dew formation. Obviously economy results from always applying just enough heat to avoid condensation according to variable weather conditions. This can be accomplished by automatically varying the temperature maintained within the building. We are not concerned with actual relative humidity of the air but only in keeping the air above its dew-point temperature when it comes in contact with the walls, switchboards and so forth. Consequently control may be based on differential temperatures rather than on relative humidity alone.

This control by differential temperatures can be accomplished by a dual instrument known as a "dew-point control" which is mounted on the inside wall of the room. It has a temperature sensitive bulb exposed to the internal air, and it also has a capillary tube extending through a hole in the wall with a sensitive bulb mounted outdoors where it is responsive to outdoor temperature. These two temperature sensitive elements apply pressure to a common lever arm in the instrument so that a resultant movement of this lever occurs and varies the actual temperature adjustment at which the instrument operates to turn on or turn off the heat. The instrument is calibrated to maintain an indoor temperature of approximately 15 F higher than the outdoor temperature within the range of 35 F inside to 65 F inside. Below 35 F inside, the instrument will always call for heat regardless of the outdoor temperature and above 65 F inside, the instrument will always shut off heat regardless of the outdoor temperature. It will thus be seen that in weather of 25 F outside, the control will maintain a constant inside temperature of 40 F, whereas with a temperature of 50 F outside, a temperature of 65 F will be maintained inside.

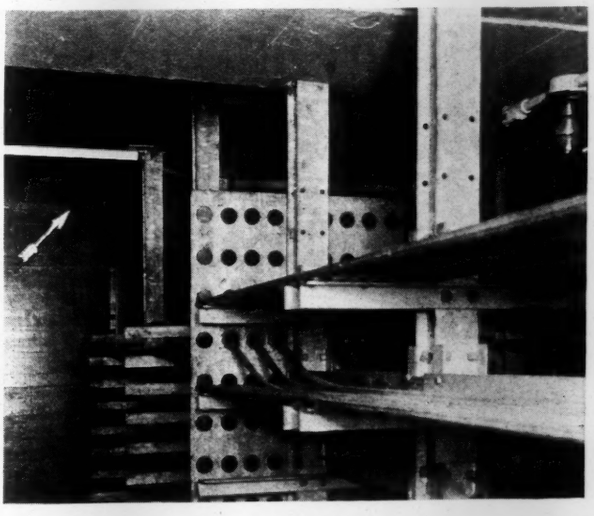
This means that at 25 F outdoors, a dew-point controlled heating system will use only enough heat to balance losses based on a 15 F temperature difference between indoors and outdoors. With the ordinary constant temperature type of thermostatic control set at 65 F, enough heat would be used under similar weather conditions to balance the heat losses at a temperature difference of 40 F, or 2.66 times as much heat energy as the variable dew-point control system would require. Both would be equally effective in preventing condensation because as long as the indoor temperature is maintained at 15 F warmer than the outdoor temperature, any infiltration of outdoor air will strike surfaces that are warmer instead of cooler than the incoming air and its dew-point temperature will not be reached, hence there will be no condensation, or dew formation. This system has been successfully used at damp seashore locations where submarine telegraph cables join the land wires. It has also been effective in electric power substations around the Great Lakes, particularly near Cleveland where a very high humidity is frequently encountered; also for high voltage cable tunnels in the Coast region of the Pacific Northwest. Many electric power stations, control rooms, meter compartments and so forth are protected by this method throughout all parts of the country.

To provide for the comfort of men temporarily working in such unattended places, it is customary to provide a bypass switch normally locked in a position to maintain the differential dew-point control, but arranged so that an authorized man can set the switch to a bypass position which will permit temporarily bringing the temperature up to the desired comfort point. Steam is seldom available for heating buildings of this kind but any other form of automatic heating system can be used. A thorough circulation of the warmed air by fans having good outlet velocities is necessary to equalize temperatures and humidity. Usually electricity is the

Left—A power station office heated by multi-blade type electric blower heaters set flush in the wall behind grille doors. Thermostat controlling the temperature is seen at the left of the door



Right—A cable tunnel at the Ariel Dam in which condensation is prevented by fan circulation electric unit heaters governed by a dewpoint control system. The arrow points to the dewpoint control instrument



most convenient and practical source of heat energy and requires a minimum of space and attention.

Drying Out Electrical Equipment

Whenever electrical machinery has been flooded or has absorbed dampness from standing unused in a moist atmosphere, it is very necessary to dry it out before it

Close-up showing sectional electric heaters being assembled and lowered into water-proof heater casings suspended in wells in the masonry at Safe Harbor Dam

is again placed in operation. Sometimes special heating equipment is built into apparatus for this purpose. In other cases, stationary or portable blower heaters are installed to blow warm air over the equipment. As an example—in the unwatering rooms at the base of the Conowingo Dam, the pump motors are protected by special blower type electric heaters which discharge hot air over them as required to keep them in a dry and safe operating condition. Similar methods are used for protecting motors in pumping stations of many kinds. In other cases, large portable blower heaters are used, being wheeled into position to dry out electrical generators, motors, switchboards, or cable tunnels by blasts of warm air. Such portable equipment is also useful during construction work for drying out damp rooms and for the comfort of workmen.

Moisture a Menace to Insulating Oils

High voltage switches and transformers contain special refined oil as an insulating medium. This oil tends to absorb moisture and every possible precaution must be taken to avoid contamination, as even a very small amount of moisture greatly reduces the dielectric value. Large power plants and substations have elaborate systems for maintaining the dielectric quality of their insulating oils. Constant tests are made and at the first sign of deterioration, the unsafe oil is pumped to storage reservoirs and replaced with purified oil. Used oil is carefully purified to bring it back to its proper standard as an insulator. From the storage tanks it is pumped through heaters and carefully raised to temperature of 150 to 180 F. It is then re-distilled or treated with a centrifuge and filter. After purification it is stored in tanks protected by mercury seal anti-breathers which maintain approximately atmospheric pressure regardless of the oil level but prevent the entrance of water vapor.

As oils are composed of hydrocarbon compounds it is very important to avoid overheating them because any decomposition produces particles of solid carbon, and these are electrical conductors. Hence, heating equipment must be designed with large surfaces of low density heat input and a positive circulation of the oil over these heating surfaces must be maintained. Accurate automatic temperature regulation is necessary and electric heat is generally used because of the accuracy with which it can be controlled. Oil purifying rooms must be heated to a comfortable temperature for operators and reasonable ventilation provided because a certain amount of oil vapor is frequently given off during purification. Fan circulation unit heaters are generally best for this purpose. Filter presses contain a large number of filter papers and these have to be frequently cleaned and dried. This drying is done on racks in electrically heated ovens with automatic temperature control to prevent charring the papers.

Oil filled cables for high voltage power transmission are a comparatively recent development. The conductor is enclosed in a sealed metal tubing which is filled under pressure with insulating oil of high dielectric quality. This oil serves as an insulator and also helps conduct heat away from the cable. Oil is prepared for this exacting service by a vacuum distillation process which removes dissolved air, gases, resins, polymerized compounds, acids, moisture and other impurities. Before entering the vacuum chamber, the oil is heated to 215 F in a circulating type of low density electric heater.

These are simply some typical examples of how the use of heat helps in carrying out the complex operations of the electrical industry. Moisture and cold are familiar adversaries in every branch of engineering work and it is to be hoped this article may help suggest methods for solving special problems in other fields.

Temperature and Humidity Requirements for Industrial Process Air Conditioning*

Product	Process	Temp. Deg Fahr	Relative Humidity Per Cent
Cotton	Carding	68-73	50
	Combing	68-73	60-65
	Roving	68-73	50-60
	Spinning	68-73	60-65
	Spooling, Twisting	68-73	65
	Warping	68-73	65
	Weaving	68-73	75-80
Wool	Carding	73-77	65-70
	Spinning	73-77	55-60
	Weaving	68-73	50-55
	Storage for Shipping	68-73	55-60
Silk	Dressing	69-77	60-65
	Spinning	69-77	65-70
	Throwing	69-77	65-70
	Weaving	69-77	60-70
Rayon	Filament Mfg.†	———	———
	Winding	70-80	40-50
	Weaving	80-85	55-60
Confectionery	Chocolate Enrobing	64	55
	Hard Candy Making	69	50
	Storage	30-59	70-55
Tobacco	Softening	84	85
	Cigar and Cigarette Making	69-73	55-70
Printing	Lithographing	69	45
	Relief and Offset	77	45
	Folding	77	65
	Binding	69	45
Baking	Dough—Fermentation	80	65
	Proofing	89-95	80-90
	Loaf Cooling	69	65
Electrical Cable	Winding Insulation	104	5
Cellulose Lacquers	Application	75	20
Munitions	Fuse Loading	69	55
Cereals	Seal Packing		
	Prepared Crisp Cereals	73	45.5

*Courtesy, The Foxboro Company, Foxboro, Mass.
†Varies according to process, fibre and product

Bush Terminal Company Proves Wisdom of

MODERNIZING Plant & Building HEATING SYSTEMS

New System Saves 40% of Steam, Cuts Maintenance Costs

By **Fred L. A. Schmidt***

TO THE MASS of evidence proving the soundness of expenditures for modernizing obsolete heating systems in order to reduce plant and building operating costs is added the experience of the Bush Terminal Company, which is now saving 40 per cent of former heating steam consumption, saved 959 tons of coal in April 1934 over April 1933, has cut maintenance costs. A valuable lesson is to be learned from the story of this modernization project told in this outstanding article.

THE Bush Terminal Company, located on the New York Bay waterfront in Brooklyn, N. Y., is a vast shipping and industrial enterprise which provides complete warehousing, manufacturing and distribution facilities. To eight great piers, ships from all quarters of the globe bring their cargos of raw materials, sugar, cocoa, chicle, pulp, and hundreds of other products. Numerous warehouses store these goods until re-shipped or used by manufacturers in the Bush model loft buildings to produce goods for domestic consumption. Eighteen loft buildings house scores of manufacturing plants which depend to a large extent upon these raw materials, and whose finished products cover a widely diversified range. The entire enterprise could easily support a city of 150,000 inhabitants.

*Consulting Engineer, Brooklyn, N. Y.

Modernization Saves 40% in Steam for Heating

It is our purpose here to describe the modernization of the heating system of the Terminal, one of the outstanding jobs of this kind and one which drives home in forceful manner the soundness of expenditures for reducing plant and building heating costs. Some idea of the heating load may be gained from the fact that the sixteen model lofts included in the remodeling program have installed over 616,000 sq ft of radiation, the heating steam load with full pressure applied to the system being approximately 185,000 pounds per hour. Under normal operating conditions the steam required by the tenants of the Terminal Company as well as the steam for heating is supplied at 120 lb pressure from what is known as the 41st St. boiler plant which is equipped with four 1000 horsepower boilers supplemented by a boiler house at 33rd St. which is used for peak loads and emergencies.

The savings which the completed modernization proj-

The Bush Terminal has been said to be "the greatest consolidation in the world of every modern facility for the manufacture, storage and distribution of manufactured goods and merchandise under conditions of maximum economy, efficiency and safety.' It covers 200 acres fully developed and has for years been accepted as a model for industrial and terminal developments.

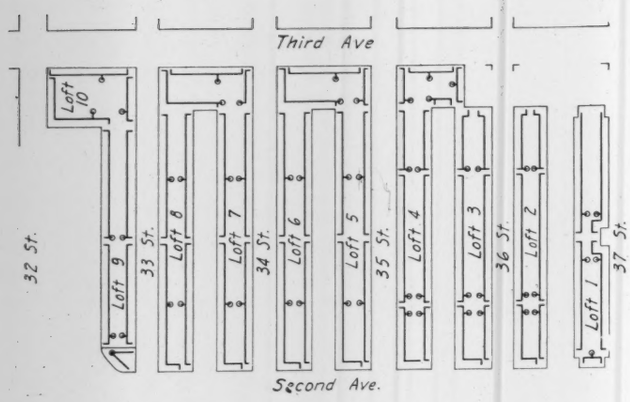
There are eighteen model industrial loft buildings, all of concrete and steel except one which is of brick and steel. It was in Bush Terminal that the first all concrete building was constructed in this country. These buildings have an aggregate gross floor area of approximately 6,000,000 sq ft and an aggregate volume of approximately 82½ million cu ft.

The buildings are equipped with up-to-date facilities, the walls being 80 per cent glass, designed to afford maximum light and air necessary to the well being and efficiency of employees. Space may be had in a wide variety of types, according to the requirements of the manufacturer or distributor. Units of 5,000 to 140,000 sq ft are available on one floor. The lay-out is such that there is space with light on one, two or three sides, also areas with light on three sides and large skylights. All units are served by elevators at each end. The elevator capacity per square foot of space is in excess of that found in many of the latest office buildings.

The buildings are 100 per cent sprinkled and the fire record which was established over a period of many years is equalled by but few, if any industrial sections in the country. High pressure steam and electric current are furnished by the Company and are available throughout the twenty-four hours.

Each building is served by railroad tracks which connect with all of the roads reaching New York. The freight elevators are in effect union freight stations for the receipt and delivery of tenants' shipments. Shipments moving via rail are handled by the Company between freight cars and tenants' space. There are also special arrangements for the handling of waterline shipments, both domestic and foreign.

The Company's operations may be described by referring to these buildings as "industrial apartment houses" as the theory of their construction and operation is identically the same as that which resulted in the development of apartment houses. The warehouse activity includes some 125 buildings offering three types of service: general warehousing, distribution and cold storage. The general warehouses handle largely raw products such as sugar, coffee, tea, rubber, cotton, tobacco,



ect will make are indicated by the actual fuel savings for the month of April, 1934, at which time twelve model loft buildings were complete and operating under the new zone control system.

During April, 1933, a total of 3866 tons of coal were required for all purposes while the month of April, 1934 (with approximately the same number of degree days) required 2907 tons, a net saving of 959 tons or 24 per cent of the total fuel for all purposes. The heating steam averages approximately 60 per cent of the total steam used so that the actual saving in steam required for heating is about 40 per cent. Approximately one-half of this saving is due to the change from single pipe to two pipe operation and elimination of leakage wastes. The other half (about 20 to 25 per cent of the total heating steam) is due to controlled operation of the new system. With the completion of the 16 buildings, a slightly larger saving is anticipated.

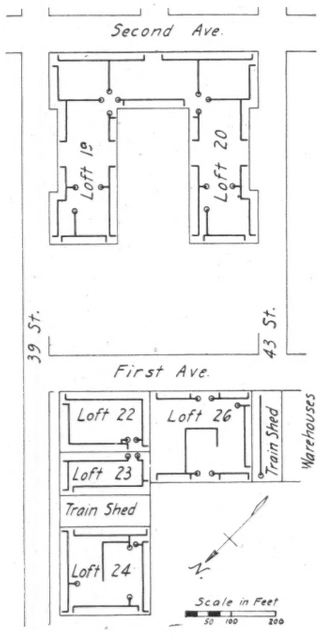
An additional saving (which cannot be evaluated) is in maintenance; 6700 sets of check valves and air valves have been removed together with steam pumps, air ejectors and appurtenances. All defective piping has been replaced. The operation has been centralized so that a single operator can control each of the 85 zones in a few minutes.

The gross savings resulting from the rebuilding of these heating systems make the investment very attractive. It is obvious from the experience of the Bush Terminal Company that plant and building owners and operators who are seeking to reduce costs can well afford to overhaul their present heating systems and install modern equipment to control them.

The System Before Modernization

The model loft buildings were equipped with one pipe downfeed Paul steam heating systems, the returns being brought to condensate tanks in the basement of each building from which the condensate was returned to the boiler plant by means of float controlled reciprocating steam pumps. The radiators were of standard return bend pipe coil construction, each coil being equipped with

Plan of the various buildings showing the arrangement of the zones, the piping, and the regulators



a gate valve in the steam supply runout, a check valve in the return runout, with an air valve between the check valve and the coil. The later model loft buildings had similar systems except that vacuum air valves were installed with a vacuum return line and steam operated air ejectors.

The pressure of the systems was controlled by hand adjusted pressure regulators which reduced the pressure from 100 lb at the distribution mains to 5 lb on the

---

etc. The distribution division specializes in the carrying of "spot stocks" of manufactured products for companies located throughout the country and the distributing of same, not only within the New York metropolitan district, but in many instances throughout the east generally. This division serves as the branch house or stock room of the manufacturer. The Bush Terminal Company, in connection with this activity, performs all of the handling and detail and in some instances where required by the manufacturer, does invoicing. The Company does not do any selling or merchandising as such but contents itself with the rendering of a specialized service the use of which by out-of-town manufacturers has grown steadily over a period of ten years or more.

The cold storage plant renders much the same service as that of the distribution division but handles, of course, only those commodities which require controlled and low temperatures. The general warehouses contain approximately 26,000,000 cu ft of space—the cold storage plant in the neighborhood of 1,000,000 cu ft and the distribution division several hundred thousand square feet.

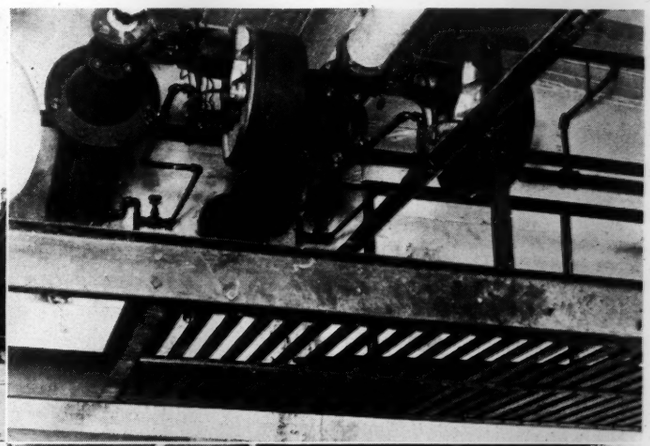
There are eight piers containing slightly less than 2,000,000 sq ft of floor space. The piers are approximately 1,300 ft long and range from 150 to 272 ft wide. One of them is double-decked and is the largest double-decked, covered pier in the world. It is possible to berth and work 35 average freight steamers at one and the same time. The piers are reached by all of the New York railroads and freight cars may be placed on the piers for loading or unloading. They are leased by the Company to steamship lines operating services to virtually all parts of the world, including Pacific Coast ports.

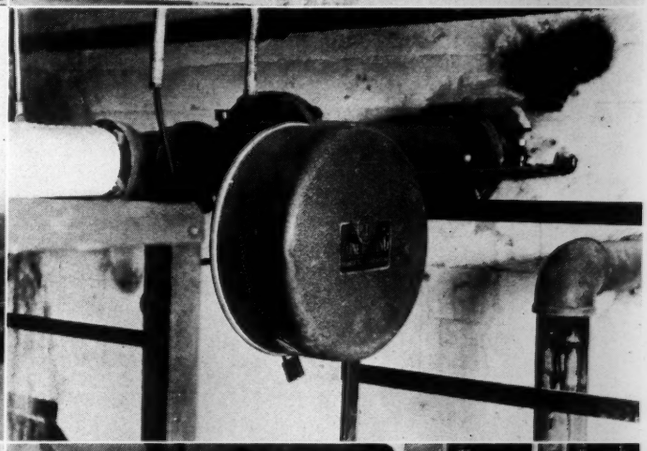
Within the confines of the Terminal there is a plant for the fumigation or sterilization of such commodities as cotton, wool, waste, chestnuts,, etc.

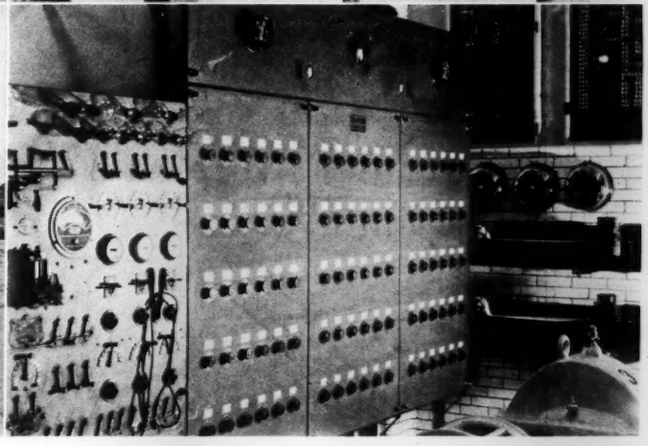
The sprinkler system which is both gravity and pressure, is the largest individual sprinkler unit in the country. It protects, in addition to the loft buildings, every other structure in the plant.

The plant as a whole is served by some 35 miles of railroad track owned and operated by the Company which also owns and operates carfloats and tugs for the handling of traffic to and from the rail heads of the New York trunk line railroads. The tracks mentioned reach each building in Bush Terminal so that the necessity for trucking is reduced to a minimum.

The organization also owns and operates a large fleet of motor trucks which covers daily the entire metropolitan area. This service is maintained in the interest of those doing business in the Terminal as the Company does not engage in the general trucking business.







Photographs Courtesy of Webster Tallmadge & Co., Inc., N. Y.

LEFT—TOP TO BOTTOM

Looking east on 36th St. toward 3rd Ave.; 2nd Ave. in foreground showing railroad facilities between Buildings Nos. 3 and 4—characteristic arrangement. The railroad connects all warehouses, piers and car-float bridges, etc. The radio towers are the ones that caught the Titanic's call for help; it was an experimental station at that time

Buildings 2 to 9 inclusive are identical. Building 10 is shorter and taller. It is over a mile and a quarter between this building (No. 10) and the control board in the pumping station from which the heat is controlled

Buildings 19 and 20 cover practically one city block each and together have approximately 100,000 sq ft E. D. R. This view is looking S. S. E. and shows the vast exposure to cold winds from northwest directly off the open 5-mile stretch across New York Harbor. The heating system has withstood severe weather of minus 15 degrees Fahrenheit and a 20 mile N. W. wind

RIGHT—TOP TO BOTTOM

Zone heating regulators in a cramped location—the stair hall at top floor of Building No. 20. They receive 120 lb steam from the top of a 4 in. high pressure riser supplying steam also to industrial tenants on the way up. The low pressure side of the regulators receives such exhaust steam as available. The regulators automatically control the make-up

All 85 of the regulators are standardized 2 in. with identical roughing-in dimensions

The zone heating centralized remote control board located in the fire pumping station. The upper part is fitted with outside temperature indicators for shady side and sunny side; also indicators for wind direction and speed. The small dials are meters and switches for remote control circuit. There are 85 controls for 85 zones and 5 spares for future

system. These valves—located on the top floors of the buildings—required the services of several men to change pressure settings each time the weather conditions made it necessary.

Economic Life of a Heating System

The economic life of a heating system has been lived when the difference in operating cost between the existing system and a new one can be capitalized favorably with the cost of a new system. Frequently a new installation becomes a very attractive investment. Operating costs are principally steam (fuel) and maintenance, both of which increase rapidly with the age of the system. When maintenance is deferred the steam cost increases at a rate greater than normal. Maintenance costs increase because of corrosion of pipes, erosion of valves, worn-out air valves, checks, traps, pumps and other conditions which affect the ability to control the steam supply. *During recent years the development of zoning, temperature controls, gravity flow atmospheric pressure returns and other steam saving features has made obsolescence a greater factor than depreciation in capitalizing improvements.*

The experience with the Bush Terminal systems indicates that the economic life of a heating system from the standpoint of depreciation ranges from 15 to 18 years. The engineering survey indicated the advisability of replacing all worn out piping and installing a method of controlling the heating systems from a central point.

The System as Modernized

The method adopted as the most suitable for meeting particular requirements of this job is a system of zoned orifice control in which an accurately calibrated orifice is used in the inlet of each radiator, usually in the valve union. When the full system steam pressure of 3 lb is applied, this orifice will permit only enough steam to enter the radiator to make it 100 per cent hot; that is, the maximum amount of steam which it will condense. No traps, air valves, checks or other devices to prevent steam from flowing through the radiator into the return lines are used.

The steam supply to each and any zone of the system is adjustable at will by means of remote controlled regulating valves which permit operation from no steam, *i. e.* valve closed, increasing to the maximum as required by the weather. An accurate pressure differential across the orifices is obtained for any desired amount of heat because the return mains are vented, maintaining the radiators and all returns at atmospheric pressure, which is practically constant. The regulators reduce the pressure from 100 lb on the mains to system pressure without the use of intermediate devices.

The complete project contemplated the installation of the new system in 16 of the 18 buildings, using 85 zones. The arrangement of zones was governed by the normal sun and wind exposures, type of occupancy, flexibility of operation and capitalized cost. A garage was made a zone because heat was required only during freezing weather and then only to prevent freezing. Another tenant required full heat at times when heat could be reduced in adjacent space. The amount of radiation in the zones ranges from 1050 to 14,800 sq ft.

The original layout of the steam piping made the mechanics of zoning a very simple matter. The old regulating valves were removed and, where necessary, short lengths of pipe were rerouted to accommodate the installation of zone control valves. A 30-in. spool piece was installed on the supply side of the valve to permit future installation of flow meters. In certain instances as many as four zone control valves were installed to replace a single pressure regulator. The valves are operated by remote electric control from the fire pump house located more than a mile from the furthest control valve.

The control board is made of ebony asbestos and has mounted on it 85 transmitters, one for each control valve, space for 15 future transmitters, two outdoor temperature indicators—one on the sunny side and one on the shady side of the building—and a wind indicator which gives both direction and speed of the wind. Direct current for operating the control circuits is obtained with a rectifier connected to the lighting circuit and alternating current for operating the control valve motors is obtained from the lighting panel nearest the valve. Standard lead covered multi-conductor telephone cable is used for the electrical control circuits, the general construction being quite similar to that used in running telephone circuits, but two wires are used for each control valve, the third connection being the ground. A total of 210 miles of wire was required for the control systems.

The dry return systems in the lofts are new. The sizes of the risers and mains are smaller than those used in vacuum return installations because the dry returns leave the radiators at atmospheric pressure and a temperature which does not exceed 140 F, at 100 per cent capacity, the air being purged from the returns by the use of atmospheric vents. It was possible to salvage a large portion of the old wet return system, the zones being isolated from each other by installing a float trap in the heel of each steam riser. The greatest amount of work required was the replacement of worn out air lines; had these been in good condition they could have been used for the new dry returns, but many were plugged with iron oxide or otherwise fouled. The new gravity air return risers were made ¾ pipe to fit in the old floor sleeves from which the air lines had been removed. These risers were found to be more than ample.

160,000 Feet of Pipe Used

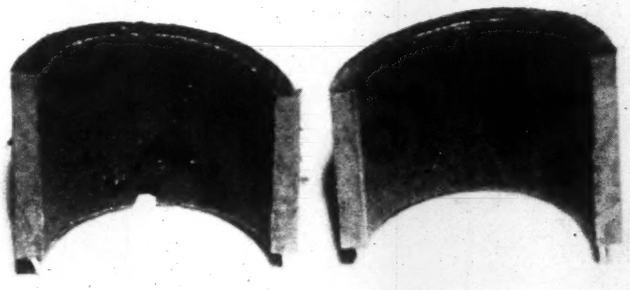
Of the 6700 pipe coils originally installed, 1200 have been replaced with other forms of radiation. The depreciation of the coils was due mainly to corrosion on the outer surface and erosion in the inner surface at the point where the condensate leaves the end of the pipe. Ten of the loft buildings occupy adjacent space in the area bounded by the 32nd St., 37th St., Second and Third Aves. Each of these had been equipped with a condensate tank and a separate pump. A new return main was installed which permits the condensate to flow by gravity from the return mains in each building to a central tank at which point a float operated electrically driven pump sends it back to the 41st St. boiler plant. This change made it possible to discontinue ten reciprocating steam pumps with their auxiliaries.

The extent of the work made it desirable to centralize operations. A complete pipe shop was installed together with a stock room, and delivery service to provide a

continuous flow of fittings and cut pipe to the job. The shop handled a total of over 160,000 feet of pipe, over 50 per cent of which was ¾ in, and about 52,000 ready-cut nipples. It produced 25,000 special lengths of pipe and handled over 60,000 fittings in the course of nine months' operation. The ready-cut nipples were obtained in standard lengths ranging from 1 to 6 in. in ½ in. sizes. Longer pieces were cut to measure in the shop—usually in quantity as methods were devised to use "standard" lengths in a large portion of the work.

A cost system was used to keep a check on the daily production and to determine to what extent the field conditions in the various lofts affected the cost.

By **H. A. Ward***



A high pressure line trapped into this return. Top half of the pipe (left) was found to be deeply pitted. The bottom half of the pipe (right) was clean

Corrosion of Piping

—an experience report

CORROSION is one of those problems which afflict the heating, piping and air conditioning engineer and about which too much will never be known. The writer does not recall seeing any published information on the corrosion of ferrous piping completely filled by high temperature water; for this reason our experiences with corrosion are briefly outlined here with the hope that they will be of aid and interest to others.

After twenty years service in our central heating system, a 6-inch underground gravity return line so installed that the exterior of the pipe was kept dry was found to be as good as new; in this case the pipe was never completely filled with water. A similar pump return filled with water punctured within five years wherever a drip from a high pressure steam line trapped into it without first passing through radiators to cool the trap discharge down to near that of the return water. The punctures developed diametrically opposite the trap connection, a large pit 2 inches in diameter forming and finally puncturing through at its center.

Just outside the central heating plant a 2-inch extra strong pump-return line to a nearby building for several feet is installed in the open. Just inside the plant a high pressure line is trapped into the return without being cooled down. At the end of four years a leak developed on the top of the return just outside the plant. This was temporarily repaired with a wooden plug. Within three months a half dozen punctures were so repaired. When finally replaced the top half of this pipe was found deeply pitted over a fair percentage of its area, the sides of the pits being vertical. The bottom half of the pipe was clean (see cut).

Originally we had two central heating plants performing similar services under like conditions. The life of the feed water lines from pumps to boilers in one of the plants was indefinitely long. In the other plant the lines had to be renewed every two or three years. In both cases, gravity heating returns from outside buildings discharged into the upper part of 4 ft x 20 ft surge tanks. The minimum level of water was maintained by a float valve controlling the make-up water from the city service. In the first case the make-up water entered the tank with the returns. In the second case it entered the bottom of the tank through an independent opening.

We checked up on the oxygen content of water at these stations. The water from the pumps or tanks at the first plant gave a small amount of oxygen. At the second plant the water from upper levels of the tank tested oxygen free but water from the bottom of the same tank was milky with air bubbles.

At the first plant the air in the make-up water was fairly well eliminated by mixing with the high temperature return water or first coming in contact

*Department of Buildings and Grounds, Cornell University, Ithaca, N. Y.

with the hot water as it cascaded into the tank. At the second plant, much of the air in the make-up water was freed by the increased temperature but being blanketed by several inches of hot water did not escape but was carried into the pump lines in a very active state. The bottom of the surge tank was also badly pitted.

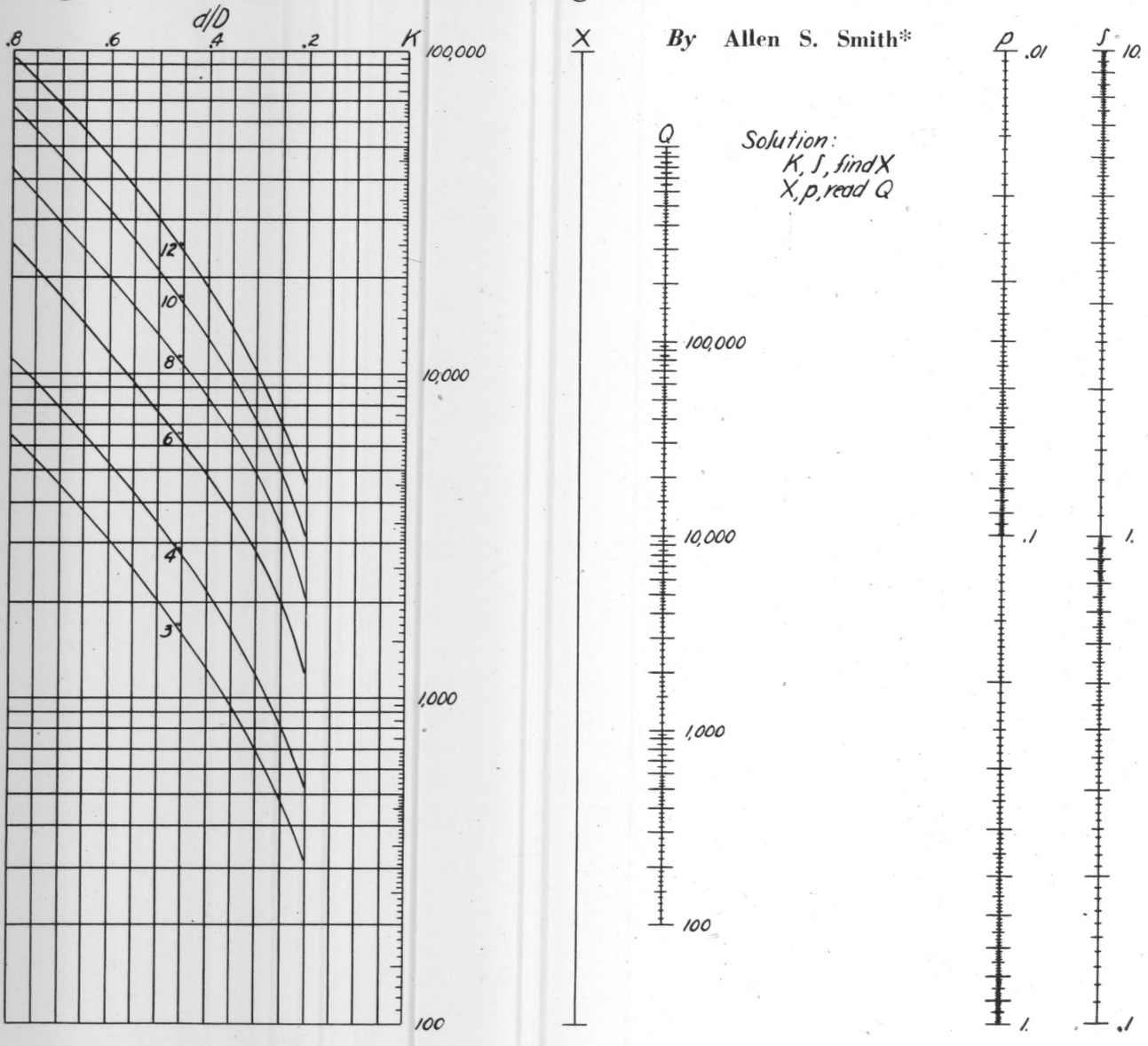
Conditions at the present new central heating plant duplicate those of the first plant. Recent examinations of the surge tanks (which have been in service several years) show them to be free from corrosion.

Any marked temperature increase in water saturated with air releases some of the air. When the released air cannot escape from the water as in the case of a line completely filled with water, at high temperatures it becomes very active.

Tap water is usually saturated with air. Introduction of tap water to hot return lines filled with water is equivalent to adding a weak but active acid.

Vacuum pumps for heating systems discharge quantities of air mixed with discharged water. Unless provision is made for separating this air from the water after leaving the pump, corrosion in the discharge pipe will be active.

Alignment Chart for Solving Problems in Orifice Flow



By **Allen S. Smith***

A SLIDE rule for solving problems in flow through orifices was devised some time ago by H. G. Schnetzler.[1] His calculations were based on the general formula

$$Q = c \frac{\pi}{4} d^2 \sqrt{2gh} \quad \ldots\ldots (1)$$

using values for the orifice coefficient c determined with the downstream pressure tap located at 4.5 pipe diameters from the orifice, *i. e.* at the point of maximum restoration of pressure. He stated that it was more convenient to locate the downstream pressure tap at this point than at the *vena contracta* because the latter shifts with the orifice ratio and the location of the pressure tap need not be changed when a change is made in the ratio.

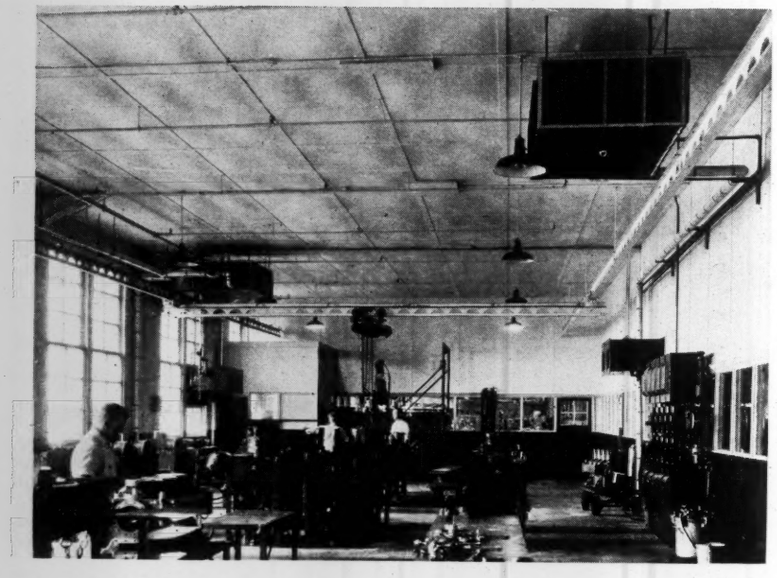
[1] *Ind. Eng. Chem.*, 19, 1027 (1927).
*Amarillo, Texas.

The location of the downstream pressure at 4.5 diameters from the orifice is seldom used in practice except for the measurement of natural gas because long runs of straight pipe are essential on both sides of the orifice. When the velocity is very great, it may be necessary to use this location to keep the area of the orifice within 70 per cent of the area of the pipe section. "The results so obtained are not as reliable as those determined at the *vena contracta* because the condition of the pipe surface has an effect on the restoration of pressure."[2]

To facilitate the computation of the rate of flow of fluids through orifices, or the selection of an orifice of proper size for a given flow, particularly in experimental work, an alignment chart has been constructed which materially reduces the labor involved in these calculations. The chart may be used with a precision of about —2 per cent. The equation which has been used is the same as that derived by Schnetzler,

$$Q = K\sqrt{\frac{l}{\rho}} \qquad \ldots\ldots\ldots\

By J. W. Baugher, Jr.*



Three suspended air-conditioning units circulate cool air in the room. Two of the three units are shown in this view

Air Conditioning System

Does Two Jobs

—Protects Machined Parts, Improves Working Conditions

IMPROVING manufacturing conditions and maintaining a comfortable and beneficial atmosphere for employees is the dual function of an air-conditioning system which serves a room at our East Pittsburgh Works. This room is used for the assembly and running-in of air-conditioning compressors, and is closed off completely from the rest of the shop aisle to prevent any dirt from the adjoining machining section from getting on the accurately-machined parts of the compressors. In order to accomplish this it was decided to maintain a pressure within the room by means of fans so that all the leakage would be outward, and at the same time bring all of the air in through filters. It was appreciated, however, that during the summer months the heat liberated in the room plus that transmitted from the outside would make the temperature unbearable and would greatly reduce the efficiency of the workmen, unless the room were cooled.

Cooling Load Totals 18 Tons

The room (which is 102 ft long, 30 ft wide and has a 15 ft ceiling) is enclosed on three sides by partitions made of two thicknesses of plaster board with an air space between, and a considerable area of glass. The fourth side is the outside wall of the building, which also has a large glass area exposed to the afternoon sun. The ceiling is made of two thicknesses of plaster board like the walls and has a skylight about 10 ft above it. The cooling load due to transmission through the walls and ceiling and to direct sun gain is consequently quite large.

The twenty-five people who work in the room under conditions of maximum production account for about seven per cent of the total load.

In one end of the room a good-sized oven, which is kept at a temperature of 125 F, supplies a considerable amount of heat; and over 40 hp in motors, most of the energy from which may be dissipated in the room, furnishes the largest single item in the cooling load.

Altogether the cooling load totals about 215,000 Btu per hr of sensible and latent heat, or nearly 18 tons of refrigeration.

Three Units Do the Job

Three suspended-type air-conditioning units were chosen to cool the room; each has a normal cooling capacity of more than 72,000 Btu per hr and delivers about 2,700 cfm of air. As much as 30 per cent of the air may be outside air brought in through a separate connection with an adjustable damper.

Both the fresh air and recirculated air are drawn into the unit through viscous-type spun-glass throw-away filters, which remove the dust and dirt. The air passes next through the direct-expansion finned-type cooling coil, where it is cooled and dehumidified. It is discharged from the air-conditioning unit by centrifugal fans at a temperature of about 60 F, falling gently to mix with the air in the room.

Freon is used as the refrigerant with an evaporating temperature in the cooling coil of about 40 F. The flow of refrigerant is controlled automatically in accordance with the cooling load by a thermostatic expansion valve.

Insulating covers prevent the transmission of heat into the unit and condensation of moisture on the outside.

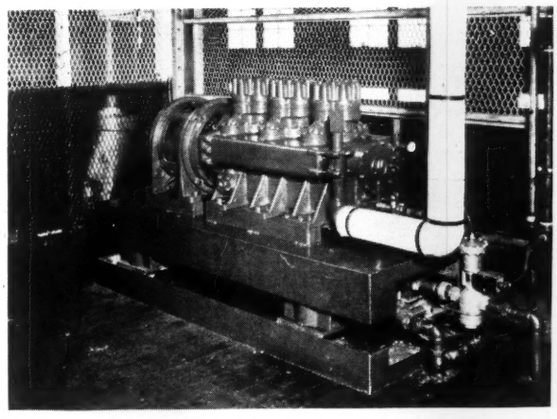
Units of this type may be obtained for winter as well as summer air conditioning by including a heating coil and a humidifier in each. Heating may be by steam or hot water with any of the common systems. Humidification is by the spray-and-target method of atomization of water, the moisture being

*Air Conditioning Engineer, Westinghouse Electric & Manufacturing Co., East Pittsburgh, Pa.

FREQUENTLY, an area in a plant must be isolated from the rest of the building for one reason or another. In this case, it was necessary to close a room used for assembly and running-in of compressors to prevent dirt from an adjoining machining section from getting on the accurately-machined parts of the compressors. A slight pressure is therefore maintained in the room so leakage will be outward, and all air is filtered. Heat liberated in the room and heat transmission from outside would make the room temperature unbearable, so cooling was also necessary. Three suspended units served by a condensing unit do the job

picked up by the air stream and further vaporized in passing through the heating coil. However, as unit heaters were already installed in the room, it was decided not to change the method of heating.

One condensing unit with a rating of 212,000 Btu per hr at a condensing pressure of 130 lb per sq in. and a suction pressure of 37 lb per sq in. is connected to all three conditioning units. The compressor is a six-cylinder machine specially designed for air conditioning operated at 1,150 rpm by a direct-connected motor. The rotor of the motor is mounted directly on an extension of the compressor crankshaft, eliminating a coupling and one motor bearing and resulting in a compact unit. Quiet operation is aided by the use of a large muffler, and freedom from vibration is insured by complete dynamic balancing and spring mounting. The condenser is water-cooled, the flow of water being regulated by an automatic valve on the unit.



The water-cooled condensing unit on the floor below is spring-mounted to isolate any vibration from the building

Location of the Equipment

The location of the air-conditioning units worked out very satisfactorily. Two units near the end of the room discharge their air toward the inside wall. These positions permit the installation of outside air connections to each of these two units by means of ducts run from the backs of the units to the upper parts of two windows. Two-and-a-half complete air changes per hour are thus provided. The third unit, which handles only recirculated air, is placed at the center of the inside wall of the room with the air delivery opposite that of the other two units.

This arrangement has resulted in excellent air distribution in the room. A gentle circulation of air free from drafts is produced, and the temperature throughout the room does not vary more than one or two degrees.

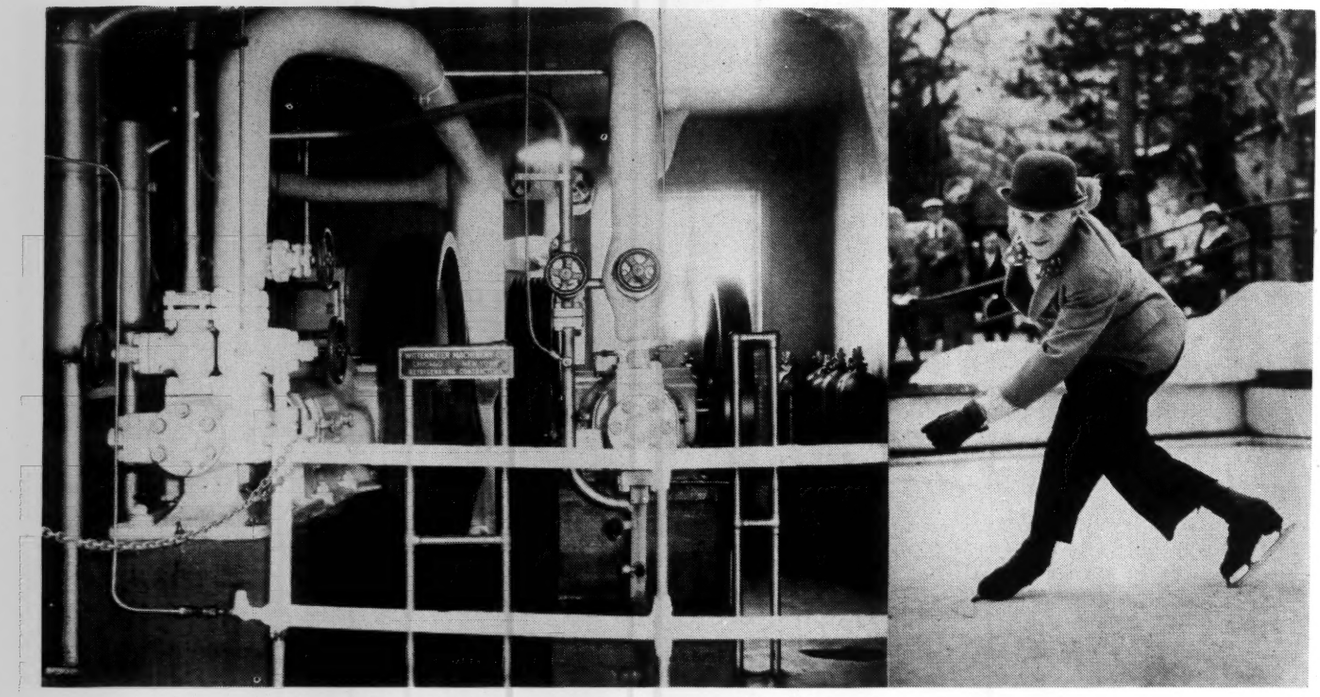
The condensing unit is located on a mezzanine floor almost directly below the air-conditioned room. The installation of refrigerant lines was therefore very much simplified, and the unit is out of the way, yet is still accessible for inspection and servicing.

The layout and installation of the piping offered few difficulties. The three air-conditioning units are operated in parallel from the condensing unit with extra heavy copper tubing carrying the refrigerant. The largest size of tubing used for the liquid line is ¾ in. and for the suction line 2½ in. The suction line is insulated to prevent the condensation of moisture on its cold surface. Valves are installed in both lines at each of the air-conditioning units so that any of them can be disconnected for servicing without making it necessary to remove the refrigerant from the entire system. Valves on the condensing unit make it possible to "pump down" the system and store the refrigerant in the liquid receiver. A ¾ in. drain line from each air-conditioning unit carries away the water removed from the air by the cooling coil. A 1¼ in. water line furnishes cooling water for the condenser.

Fan and Compressor Control

The compressor motor is operated by a magnetic starter controlled by a thermostat in the conditioned room with a normal setting about 12 F below the outside temperature. The compressor is thus started and stopped in accordance with the cooling requirements. The fans of the air-conditioning units operate continuously to provide a circulation of air in the room, even though the thermostat has stopped the condensing unit. They are subject to control, however, by magnetic starters.

The conditions in the room are most comfortable, even on the hottest, most humid days. With temperatures in the other parts of the plant of 96 F dry bulb and 84 F wet bulb (62 per cent relative humidity), the temperatures in the air-conditioned room were maintained at 82 F dry bulb and 74 F wet bulb (68 per cent relative humidity). This is a difference in dry bulb temperature of 14 F and in moisture content of 44 grains per pound of air.



Black Forest Climate Brought to Fair

Refrigerating plant at the Black Forest Village for air conditioning the restaurant and for the outdoor skating rink; signs point the way to this machine room so visitors may have a look behind the scenes. Note the frost-covered handrail. At the right, Bert Clark, one of the skating group at the Village, proves the ice is in good condition

90 Tons of Refrigeration at World's Fair Village Cools and Dehumidifies Restaurants, Freezes Outdoor Ice Skating Rink

PLANNERS of the Black Forest—one of the most popular attractions at the 1934 Chicago World's Fair—aimed to create not only a true representation of a native Black Forest village atmosphere in the appearance of the buildings, but to simulate insofar as possible the climate of that part of Germany. As investigation revealed this could be done by maintaining a temperature of 80 F and a relative humidity of 50 per cent, a 50-ton carbon dioxide refrigerating system was installed to cool and dehumidify the air for the restaurants, with a complete air change every three minutes.

In addition to the refrigeration for air conditioning, a 40-ton machine is used to freeze the 32 ft x 48 ft outdoor skating rink at the Village. For exhibition skating the ice must be not only very smooth but of greater hardness than ordinary artificially frozen ice. The exposure of the entire skating surface to the summer sun when the outdoor air temperature reaches 100 F and above demands a system capable of removing the heat projected on the ice surface by the sun's rays, the heat from the surrounding atmosphere and frequently the heat from rain drops on the ice. The 1½ in. thickness of ice is maintained by electrically welded seamless steel coils using CO_2 direct expansion. The direct expansion system is not ordinarily used on skating rinks, brine being more usual.

Ice temperatures are measured by a recording thermometer; when the temperature of the ice goes above 28 F, the ice machine is put in operation. The temperature of the ice in the skating rink is at all times under the observation of electrically actuated thermometers registering within 1/10 F.

The skating rink rests on a concrete base upon which a 4 in. layer of compressed cork insulation is laid; the sides and ends are also insulated. Between the skating performances an insulated dance floor moves over the entire ice surface. The ice is flooded four times a day which insures a smooth surface for the professional skaters appearing at the village. During a recent hot spell when the thermometer on three occasions exceeded 100 F, the rink was maintained in perfect skating condition and brought only praise from the skaters.

The total refrigerating capacity of 90 tons is installed in two units, one of 40 and the other of 50 tons capacity cross-connected so that the total 90 ton capacity can be devoted either to freezing the ice or for air cooling and conditioning purposes. The compressor motors are equipped with variable speed controls.

The condenser water is controlled and regulated by an electric water valve, water economy being an important consideration in view of the fact that the Fair management measures all the water consumed.

Heating · Piping
and Air Conditioning

American Society of Heating and Ventilating Engineers JOURNAL SECTION

1895

SEPTEMBER 1934

Table of Contents

Wind Velocities Near a Building and Their Effect on Heat Loss, by F. C. Houghten, J. L. Blackshaw, and Carl Gutberlet 399

What Is the Cooling Load Factor in Air Conditioning? by John Everetts, Jr. 405

A Method of Testing Filters Against Pollens, Bacteria and Molds, by Philip Drinker and Wm. F. Wells 408

Candidates for Membership 414

Candidates Elected 414

The material in this Section is prepared by the Publication Committee of the Society and nothing that appears in it may be reprinted, either wholly or in part, without permission. The use of the Society's name for advertising promotions is never authorized.

OFFICERS AND COUNCIL 1934

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS
51 Madison Ave., New York, N. Y.

President ..... C. V. HAYNES

First Vice-President ..... JOHN HOWATT
Second Vice-President ..... G. L. LARSON

Treasurer ..... D. S. BOYDEN
Secretary ..... A. V. HUTCHINSON

COUNCIL

C. V. HAYNES, *Chairman* — JOHN HOWATT, *Vice-Chairman*

One Year

ALBERT BUENGER, F. E. GIESECKE, W. T. JONES, J. F. MCINTIRE, W. E. STARK

Two Years

R. H. CARPENTER, J. D. CASSELL, F. C. MCINTOSH, L. W. MOON

Three Years

M. C. BEMAN, E. H. GURNEY, O. W. OTT, W. A. RUSSELL

COMMITTEES OF THE COUNCIL

Executive: JOHN HOWATT, *Chairman*
J. D. CASSELL, E. H. GURNEY

Finance: R. H. CARPENTER, *Chairman*
F. C. MCINTOSH, J. F. MCINTIRE

Meetings: W. E. STARK, *Chairman*
F. E. GIESECKE, L. W. MOON

Membership: F. C. MCINTOSH, *Chairman*
ALBERT BUENGER, W. A. RUSSELL

ADVISORY COUNCIL

W. T. Jones, *Chairman;* W. H. Carrier, Homer Addams, R. P. Bolton, S. E. Dibble, W. H. Driscoll, H. P. Gant, John F. Hale, L. A. Harding, H. M. Hart, E. Vernon Hill, J. D. Hoffman, S. A. Jellett, D. D. Kimball, S. R. Lewis, Thornton Lewis, J. I. Lyle, J. R. McColl, D. M. Quay, C. L. Riley, F. B. Rowley, F. R. Still and A. C. Willard.

COOPERATING COMMITTEES

A. S. H. V. E. representative on *National Research Council;* Prof. F. E. Giesecke (3 years).

SPECIAL COMMITTEES

Committee on Admission and Advancement: A. J. Offner, *Chairman (two years);* J. G. Eadie *(one year),* and E. N. Sanbern *(three years).*

Publication Committee: W. M. Sawdon, *Chairman;* A. P. Kratz and M. C. Beman.

Committee on Constitution and By-Laws: Thornton Lewis, *Chairman;* W. T. Jones and O. W. Ott.

Guide Publication Committee: W. L. Fleisher, *Chairman;* John Howatt, G. L. Larson, S. R. Lewis, E. N. McDonnell, W. M. Sawdon and J. H. Walker.

Committee on Ventilation Standards: W. H. Driscoll, *Chairman;* J. J. Aeberly, L. A. Harding, D. D. Kimball, J. R. McColl, C. L. Riley, W. A. Rowe, Perry West and A. C. Willard.

Committee on Code for Testing and Rating Condensation and Vacuum Pumps: John Howatt, *Chairman;* W. H. Driscoll, L. A. Harding and F. J. Linsenmeyer.

Committee on Code for Testing and Rating Convectors: R. N. Trane, *Chairman;* E. H. Beling, R. F. Connell, M. Dillman, W. Ewald, J. H. Holbrook, Hugo Hutzel, A. P. Kratz, M. G. Steele and O. G. Wendel.

NOMINATING COMMITTEE FOR 1934

Chapters	*Representative*	*Alternate*
Cincinnati	J. J. Braun	H. E. Sproull
Cleveland	C. F. Eveleth	F. A. Kitchen
Illinois	J. J. Aeberly	J. J. Hayes
Kansas City	W. A. Russell	E. K. Campbell
Massachusetts	J. F. Tuttle	W. F. Gilling
Michigan	W. G. Boales	Tom Brown
Western Michigan	S. H. Downs	J. H. Van Alsburg
Minnesota	N. D. Adams	C. E. Lewis
New York	H. W. Fiedler	V. J. Cucci
Western New York	D. J. Mahoney	J. J. Yager
Ontario	H. S. Moore	F. E. Ellis
Pacific Northwest	A. L. Pollard	Lincoln Bouillon
Philadelphia	W. R. Eichberg	M. F. Blankin
Pittsburgh	F. C. McIntosh	F. C. Houghten
St. Louis	R. J. Tenkonohy	
Southern California	L. H. Polderman	R. L. Gifford
Wisconsin	E. A. Jones	Ernest Szekely

COMMITTEE ON RESEARCH

JOHN HOWATT, *Chairman*

F. E. GIESECKE, *Vice-Chairman* — DR. A. C. WILLARD, *Technical Adviser* — F. C. HOUGHTEN, *Director*

O. P. HOOD, *Ex-Officio Member*

One Year

D. E. FRENCH, F. E. GIESECKE, L. A. HARDING, G. L. LARSON, A. P. KRATZ

Two Years

ALBERT BUENGER, S. H. DOWNS, H. N. KITCHELL, H. R. LINN, PERRY WEST

Three Years

C. A. BOOTH, E. K. CAMPBELL, JOHN HOWATT, A. J. NESBITT, J. H. WALKER

Executive Committee

JOHN HOWATT, *Chairman*

F. E. GIESECKE, H. R. LINN

Finance Committee

A. J. NESBITT, *Chairman*

N. D. ADAMS, E. C. EVANS, G. L. LARSON, J. H. WALKER

TECHNICAL ADVISORY COMMITTEES - 1934

Air Conditions and Their Relation to Living Comfort: C. P. Yaglou, *Chairman;* J. J. Aeberly, W. L. Fleisher, D. E. French, Dr. R. R. Sayers and Dr. C.-E. A. Winslow.

Air Conditioning in Treatment of Diseases: Dr. E. V. Hill, *Chairman;* N. D. Adams, J. J. Aeberly, Margaret Ingels, H. R. Linn and E. L. Stammer.

Atmospheric Dust and Air Cleaning Devices (Including Dust and Smoke): H. C. Murphy, *Chairman;* J. J. Bloomfield, M. I. Dorfan, Philip Drinker, Dr. Leonard Greenburg, S. R. Lewis, T. W. Pangborn, F. B. Rowley and Games Slayter.

Correlating Thermal Research: R. M. Conner, *Chairman;* D. S. Boyden, J. C. Fitts, H. T. Richardson and Perry West.

Corrosion: J. H. Walker, *Chairman;* H. F. Bain, E. L. Chappell, W. H. Driscoll and R. R. Seeber.

Direct and Indirect Radiation with Gravity Air Circulation: H. F. Hutzel, *Chairman;* A. P. Kratz, H. R. Linn, J. F. McIntire, J. P. Magos, T. A. Novotney, R. N. Trane and G. L. Tuve.

Gas Heating Equipment: W. E. Stark, *Chairman;* Robert Harper, E. A. Jones, Thomson King, J. F. McIntire, E. L. Tornquist and H. L. Whitelaw.

Heat Requirements of Buildings: D. S. Boyden, *Chairman;* P. D. Close, W. H. Driscoll, H. M. Hart, P. E. Holcombe, V. W. Hunter, F. B. Rowley, R. J. J. Tennant and J. H. Walker.

Heat Transfer of Finned Tubes with Forced Air Circulation: F. B. Rowley, *Chairman;* H. F. Bain, H. F. Hutzel, W. G. King, A. P. Kratz, E. J. Lindseth, L. P. Saunders, G. L. Tuve and W. E. Stark.

Minimum Temperature and Method of Introduction of Cooling Air in Classrooms: Perry West, *Chairman;* J. D. Cassell, S. R. Lewis, J. R. McColl, A. J. Nesbitt, G. E. Otis and C.-E. A Winslow.

Oil Burning Devices: H. F. Tapp, *Chairman;* Elliott Harrington, F. B. Howell, J. H. McIlvaine, L. E. Seeley and T. H. Smoot.

Pipe and Tubing (Sizes) Carrying Low Pressure Steam or Hot Water: S. R. Lewis, *Chairman;* J. C. Fitts, F. E. Giesecke, H. M. Hart, C. A. Hill, R. R. Seeber and W. K. Simpson.

Refrigeration in Relation to Air Treatment: A. P. Kratz, *Chairman;* E. A. Brandt, John Everetts, Jr., E. D. Milener, K. W. Miller, E. B. Newill, F. G. Sedgwick and J. H. Walker.

Sound in Relation to Heating and Ventilation: V. O. Knudsen, *Chairman;* Carl Ashley, C. A. Booth, F. C. McIntosh, R. F. Norris, J. S. Parkinson, C. H. Randolph, J. P. Reis and G. T. Stanton.

Ventilation of Garages and Bus Terminals: E. K. Campbell, *Chairman;* S. H. Downs, T. M. Dugan, E. C. Evans, F. H. Hecht, H. L. Moore and A. H. Sluss.

OFFICERS OF LOCAL CHAPTERS, 1933-1934

CLEVELAND
Headquarters, Cleveland
Meets: Second Thursday in Month
President, F. A. KITCHEN
1514 Prospect Ave.
Secretary, M. B. WRIGHT
Case School of Applied Science.

CINCINNATI
Headquarters, Cincinnati, O.
Meets: Second Tuesday in Month
President, H. N. KITCHELL
4528 Circle Ave.
Secretary, E. B. ROYER
6635 Iris Ave.

ILLINOIS
Headquarters, Chicago
Meets: Second Monday in Month
President, C. W. DELAND
211 N. Desplaines St.
Secretary, J. J. HAYES
53 W. Jackson Blvd.

KANSAS CITY
Headquarters, Kansas City, Mo.
Meets: Second Monday in Month
President, L. A. STEPHENSON
409 East 13th St.
Secretary, L. R. CHASE
217 Dwight Bldg.

MASSACHUSETTS
Headquarters, Boston
Meets: Second Monday in Month
President, R. S. FRANKLIN
38 Chauncy St.
Secretary, W. A. MCPHERSON
86 Dwinnell St.
West Roxbury, Mass.

MICHIGAN
Headquarters, Detroit
Meets: First Monday after the 10th of the Month
President, G. D. WINANS
2000 Second Ave.
Secretary, W. F. ARNOLDY
2847 Grand River Ave.

WESTERN MICHIGAN
Headquarters, Grand Rapids
Meets: Second Monday in Month
President, S. H. DOWNS,
211 Creston Ave.,
Kalamazoo, Mich.
Secretary, W. G. SCHLICHTING,
1417 W. Lovell St.
Kalamazoo, Mich.

MINNESOTA
Headquarters, Minneapolis
Meets: Second Monday in Month
President, A. B. ALGREN
5109 17th Ave. S.
Secretary, C. E. GAUSMAN
2360 Chilcombe, St. Paul, Minn.

NEW YORK
Headquarters, New York
Meets: Third Monday in Month
President, H. W. Fiedler
489 Fifth Ave.
Secretary, T. W. REYNOLDS
100 Pinecrest Drive,
Hastings-on-Hudson, N. Y.

WESTERN NEW YORK
Headquarters, Buffalo
Meets: Second Monday in Month
President, J. J. YAGER
425 Woodbridge Ave.
Secretary, P. S. HEDLEY
Curtiss Bldg.

ONTARIO
Headquarters, Toronto, Can.
Meets: First Monday Every Other Month
President, W. R. BLACKHALL
332 Waverley Rd.
Secretary, H. R. ROTH
1104 Bay St.

PACIFIC NORTHWEST
Headquarters, Seattle, Wash.
Meets: Second Tuesday in Month
President, A. L. POLLARD
601 Electric Bldg.
Secretary, S. D. PETERSON
473 Colman Bldg.

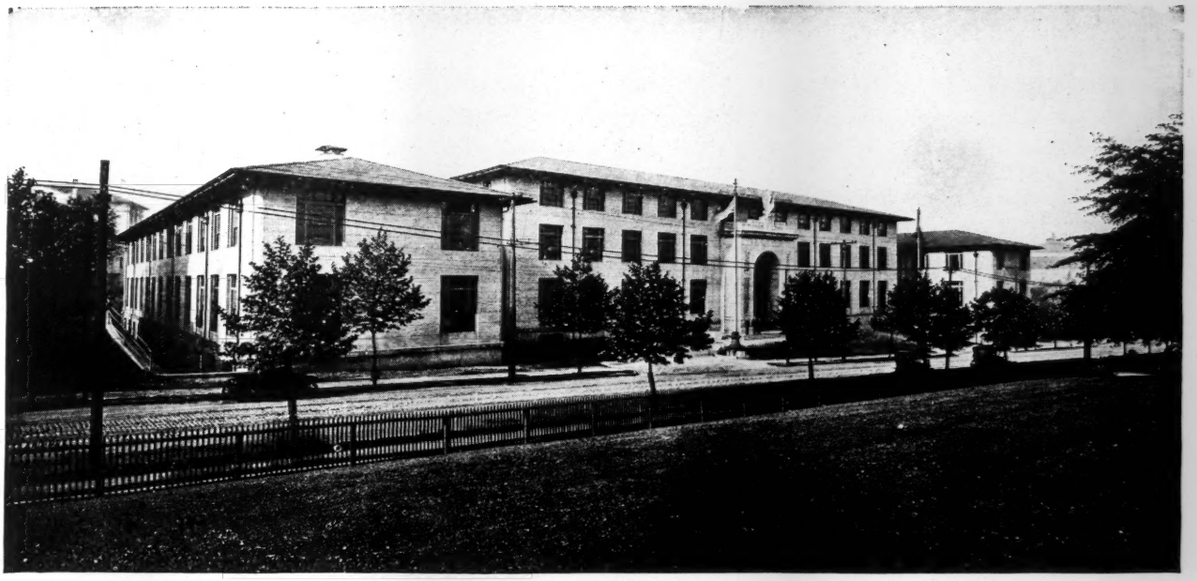
PHILADELPHIA
Headquarters, Philadelphia
Meets: Second Thursday in Month
President, W. P. Culbert
2019 Rittenhouse St.
Secretary, W. R. EICHBERG
4210 Sansom St.

PITTSBURGH
Headquarters, Pittsburgh
Meets: Second Monday in Month
President, P. A. EDWARDS
608 Wabash Bldg.
Secretary, J. L. BLACKSHAW
4800 Forbes St.

ST. LOUIS
Headquarters, St. Louis
Meets: First Wednesday in Month
President, PAUL SODEMANN
4136 Farlin Ave.
Secretary, A. L. WALTERS
7284 Richmond Pl.,
Maplewood, Mo.

SOUTHERN CALIFORNIA
Headquarters, Los Angeles
Meets: Second Tuesday in Month
President, W. H. C. NESS
1323 Channing St.
Secretary, E. H. KENDALL
309 West 12th St.

WISCONSIN
Headquarters, Milwaukee
Meets: Third Monday in Month
President, JOHN S. JUNG
1516 S. Layton Blvd.
Secretary, G. E. HOCHSTEIN
3000 W. Montana St.



Pittsburgh Experiment Station of the U. S. Bureau of Mines where the Research Laboratory of the American Society of Heating and Ventilating Engineers is located

Wind Velocities Near a Building and Their Effect on Heat Loss

By **F. C. Houghten** *, **J. L. Blackshaw** ** *(MEMBERS)*, **and Carl Gutberlet** *** *(NON-MEMBER)*

Pittsburgh, Pa.

THE estimation of the heating requirements of buildings is the most fundamental practice of the heating and ventilating engineer. It has been the subject of continuous study since the heating of buildings became an important branch of the engineering industry. The determination of the heating load of a building requires *first,* knowledge concerning heat transfer and infiltration through the various types of construction involved for a practical range of temperature differences and wind velocities; *second,* knowledge concerning the weather conditions to which the building will be subjected; *third,* knowledge concerning the proper application of the items listed in formulae to give the resultant heat loss from the completed building or room under consideration.

During recent years research has been directed more intensively toward the extension of the available data on heat transfer and infiltration, and as a result, these data are now better understood than the facts concerning the probable temperatures, wind velocities, sun effect, heat capacity, and other environmental conditions and their use in the calculation of the final results. This subject was approved as a major research project by the Committee on Research last summer, and a Technical Advisory Committee under the chairmanship of D. S. Boyden was appointed.

A great deal of consideration has since been given to plans for this study by the Committee, including the collection of data on the over-all heat demands of existing buildings, an analysis of weather bureau data for various localities with a view of determining the combined effect of wind and outside temperature, and more intensive studies of the heat loss from a few individual rooms in existing buildings. It was agreed to concentrate for the present on the third type of study, namely, investigations of individual rooms in existing buildings. Data were collected during the past heating season in buildings in a number of geographical locations, including the study here reported, which was made in eight rooms in the Grant Building, Figs. 1 and 2.

The Grant Building is a modern 40-story office building of the set-back skyscraper type built in 1926. The building is normally heated by a differential vapor system using enclosed copper convectors below all windows. Control of heat supply is obtained by varying the pressure of the steam supplied to the system from a small positive pressure in cold weather to the required vacuum in mild weather.

The rooms were chosen so as to give data on the heating demand of the different sides of the building on

*Director, **Research Engineer, ***Research Assistant, Research Laboratory, AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Pittsburgh, Pa.

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Buck Hill Falls, Pa., June, 1934.

a given floor as affected by outside weather conditions and data concerning the relation of elevation to heat demand due to exposure to wind and the chimney effect of the building. Three rooms were chosen on the 7th floor; 726 facing Third Avenue or the southwest, 707 facing Grant Street or the northwest, and 705 facing Fourth Avenue or the northeast. No typical rooms were available for test on the fourth side of the building which was devoted largely to corridors and elevator space. The

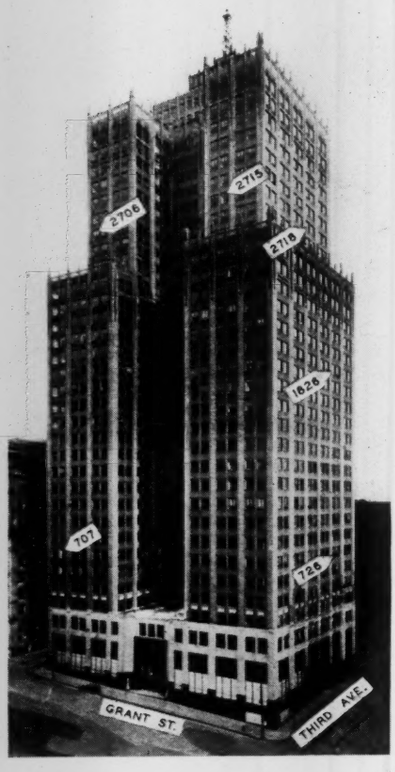


Fig. 1—The Grant Building, showing locations of six of the eight rooms used in the study

three rooms on the 7th floor were chosen to represent different exposures in the lower portion of the building, where the heating demands might be affected by other buildings in the vicinity.

Buildings across the streets which may affect wind movement near these rooms and their heights are as follows: Across Third Avenue and opposite room 726 buildings extend to the sixth floor of the Grant Building; across the corner of Third Avenue and Grant Street there is a three-story building; the lot across Grant Street near Third Avenue is vacant; across Grant Street and opposite room 707, across the corner of Grant Street and Fourth Avenue, and across Fourth Avenue and opposite room 705, buildings rise, respectively, to the levels of the 9th, 18th and 14th floors of the Grant Building. Third Avenue, Grant Street and Fourth Avenue, are respectively, 37, 88 and 58 ft wide from building to building.

A single room, 1818, was chosen on the 18th floor facing Third Avenue or the southwest, to represent conditions in the middle or the neutral zone of the building.

Four rooms were chosen on the 27th floor; 2718 facing Third Avenue or southwest, 2715 facing Grant Street or northwest, 2703 facing Fourth Avenue or northeast, and 2706 on the north corner facing both Grant Street and Fourth Avenue. These rooms were chosen to represent the different directions of exposure in the upper portion of the building. With the exception of the corner room, 2706, all rooms had but one exposure, with similarly heated offices on either side as well as above and below. The three rooms facing Third Avenue on the 7th, 18th and 27th floors were similarly located with respect to the plan of the building so as to eliminate the effect of slight variations in location on the floor.

Each room studied was equipped with electrical heaters located directly below the copper steam convector with the cabinet extended downward to 3 in. from the floor so as to enclose the electrical heater. The convection currents of air passed upwards through the steam heating unit and into the room through the same grille which is used when heating the room with steam. The arrangement of the electrical heaters with respect to the convector and its enclosure, the window and other details is shown in Fig. 3, and was observed by the use of smoke to give convection currents of air into the room of the same temperature, velocity and direction as given by normal operation of the steam system.

The electrical heaters were arranged so that a capacity of 2,560 Btu per hour was always controlled by an "on" and "off" thermostat located 36 in. above the floor in the center of the room and shielded from view of the window and convector cabinet. A simple arrangement of switching made it possible to add additional or auxiliary heating capacity in units of 640, 1,280, and 2,560 Btu per hour, according to the heating demand. This gave a very satisfactory and flexible control without the disturbing effect of throwing the entire heating load of the room on and off. Each time upon passing through the rooms the attendant adjusted the auxiliary heating capacity in an attempt to make it and that controlled by the thermostat just a little greater than sufficient to maintain the desired room temperature. All current supplied to each room for the heating units, lights and other uses passed through an integrating watt-hour meter having dials which could be read to $\pm$ 5 watt-hours or 17 Btu.

Temperatures were observed by thermocouples, at various heights between the ceiling and floor in the center of the rooms, at a single location in each of the adjoining rooms, of the surface of the ceiling and floor in the middle of the test room, the inside surface of each of the four walls of the test room, the glass surface, the outside air 3 ft away from the window, and of the outside wall surface of the building. Some of these temperatures were checked by calibrated mercury thermometers. The pressure drop through the window was observed by an inclined manometer.

Wind velocities three feet away from the building wall were observed by an anemometer held out of the window of a room adjoining each of the test rooms. At a distance of 3 ft from the window the component of the wind velocity perpendicular to the building wall and the horizontal and vertical components parallel to the building wall were observed. A cup anemometer was located on the top of the Grant Building and connected to an electrical counter on the 27th floor. Complete Government Weather Bureau data were available from the Pittsburgh station located 1320 ft away in the direction indicated in Fig. 2.

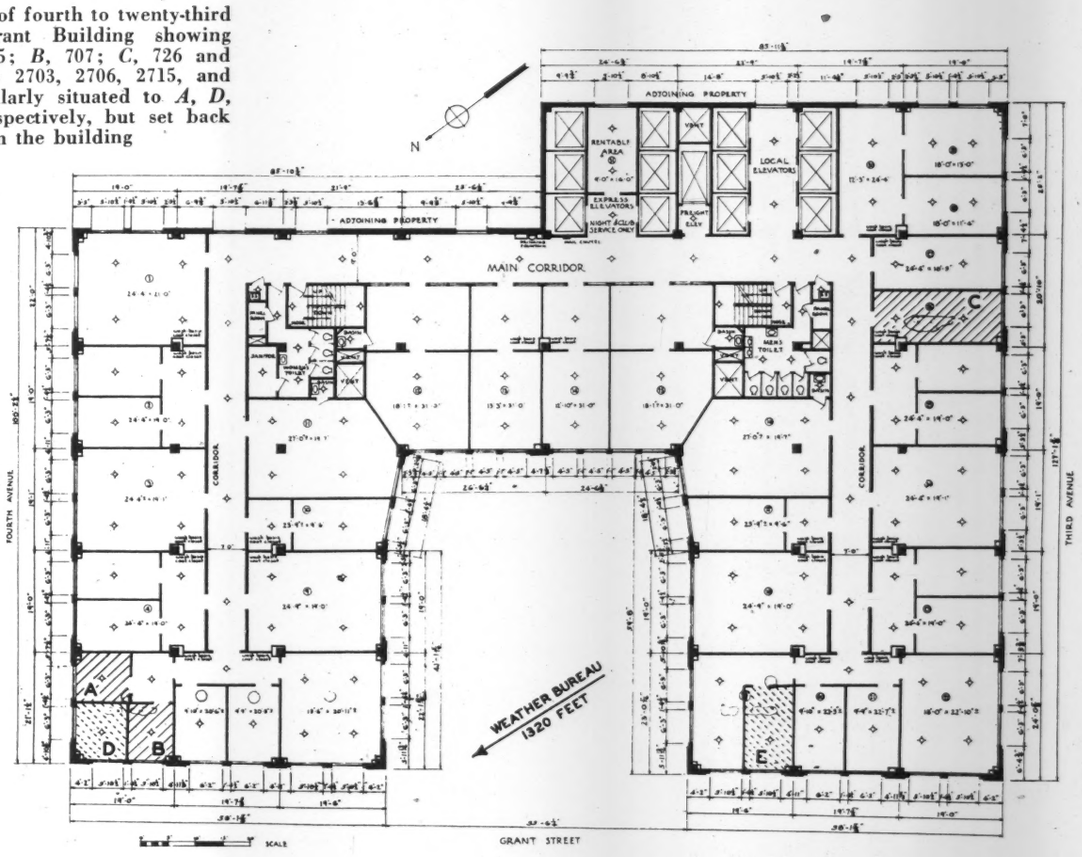


Fig. 2—Plan of fourth to twenty-third floors of Grant Building showing Rooms *A*, 705; *B*, 707; *C*, 726 and 1826. Rooms 2703, 2706, 2715, and 2718 are similarly situated to *A*, *D*, *E*, and *C*, respectively, but set back from the building

In making a test, the observer entered each room at about 8:00 A.M., turned off the steam heat which had been left on the night before and turned on the electric heating unit controlled by the thermostat and a sufficient number of auxiliary heaters so that the total was a little greater than the heat required to maintain the desired temperature when the thermostat was on. Observation of the room temperature, the temperatures of the surrounding rooms and the watt-hour meter readings were also made on this tour of the eight rooms. Frequent rounds were made of all the test rooms throughout the day, when these observations were repeated and the necessary adjustments to the temperature of the adjoining rooms or to the thermostats in the test rooms were made.

This procedure was continued until 10:00 P.M., when on the last round the observer turned off all electric connections and turned on the steam radiators before leaving for the night, thus allowing the test rooms to float through the night with the same steam heat supply and control as was maintained in the remainder of the building.

Less frequently, additional rounds were made of the test rooms when temperatures given by the thermocouples and thermometers in and about the rooms were read and the wind velocities, 3 ft outside of the window, and the pressure drops through the window were observed.

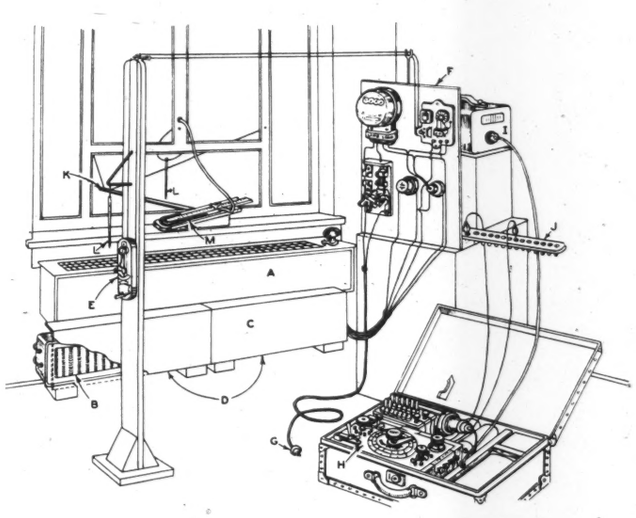


Fig. 3—Test room showing heater and test equipment

A—Original convector cabinet
B—Electric heater
C—Extended convector cabinet
D—Air inlet to cabinet
E—Thermostat at 36-in. level
F—Control panel
G—Power line
H—Precision potentiometer and auxiliary equipment arranged for portable use
I—Galvanometer
J—Selective thermocouple switch
K—Thermocouple for observing air temperature 3 ft from outside of window
L—Thermometers for observing inside and outside air temperatures
M—Inclined manometer for observing pressure difference across window

Test Results and Observations

Because of the great volume of data collected between January 18, when the tests were started, and April 12, when the last test was made, and because it was impossible to analyze any considerable number of the results while collecting the data, this report will be limited to an analysis and discussion of the results obtained from the weather bureau records, the wind velocities and temperatures observed 3 ft outside of the individual test rooms, and the temperatures of the glass surfaces. Table

1 gives wind velocities and temperatures obtained from the weather bureau records and observed near the Grant Building and results calculated therefrom for the days studied. It will be observed that the minimum daily temperatures for the period of the tests on these days range from —11 to 45 F. The maximum daily wind velocity ranged from 8 to 34 mph. It should be noted that the minimum temperature and maximum wind did not necessarily prevail at the same time.

Column L gives the total solar energy per square foot of horizontal surface for the period from sunrise to sunset as recorded by the weather bureau in Pittsburgh. Column M gives the calculated maximum rate of heat flow through a window from an inside temperature of 70 F and the most severe concurring outside temperature and wind velocity recorded by the weather bureau for the test periods. These calculations were based upon an inside glass surface film conductance coefficient, $f_i = 1.65$, and an outside coefficient, $f_0 = 1.65 + 0.232\,V$, where V is the wind velocity. This relation between the outside surface coefficient and the wind velocity is in agreement with values determined by the Laboratory in Pittsburgh[1] and by Prof. Rowley[2] at the University of Minnesota. Column N gives the time when the maximum calculated rate of heat flow occurred, which was most often in the early forenoon.

Column O gives the average rate of heat flow through the glass based upon the above considerations and the average temperature and wind velocity recorded by the weather bureau for the period of the tests.

Column P gives the wind velocity and Column Q the temperature observed 3 ft outside of the window for room 2703 at the time of the maximum calculated rate of heat flow. Column R gives the calculated rates of heat loss from a 70 F room to the outside air for the observed wind velocities and temperatures given in columns P and Q.

The rate of heat flow, H, through a window may be calculated from the difference between the inside air temperature, t_3, and the glass surface temperature, t_2, if the inside surface film conductance coefficient f_i is known, by the formula:

$$H = f_i\,(t_3 - t_2) \qquad (1)$$

Likewise, the outside film conductance coefficient f_0 is given by the formula:

$$f_0 = \frac{H}{(t_2 - t_1)} \qquad (2)$$

where t_1 = the outside temperature

or

$$f_0 = \frac{f_i\,(t_3 - t_2)}{(t_2 - t_1)} \qquad (3)$$

Column S gives the values for the outside film conductance coefficient based upon the inside room air temperature and the glass surface temperature observed at the time of the maximum calculated rate of heat flow given in column M. Column T gives the rate of heat flow through the glass based on the values of f_0 given in column S, a value of $f_i = 1.65$, an inside air temperature of 70 F, and the observed outside temperature 3 ft from the window.

Columns U, V and W give the percentages which the rates of heat flow given in columns O, R and T, respectively, are of the maximum rate of heat flow given in column M.

These percentages show that the heat flow through the window from a 70 F room calculated from the observed temperatures and wind velocities outside of the window is considerably lower than that calculated from the weather bureau data. When the rate of heat flow is calculated from the observed relation between the in-

Table 1—Test Data and Results

A	B	C	D	E	F	G	H	I	J	K	L	M	N	O	P	Q	R	S	T	U	V	W
		Weather Bureau Data													Data From Room 2703							
		Wind				Temperature Fahrenheit						Heat Flow Btu/Sq Ft/Hour Through Glass								Percentage		
			Velocity-MPH																			
Test Number	Date — 1934	Direction	Average	Maximum	Minimum	Average	Maximum	Minimum During Test Period	Min. From 10:00PM To Following 10:00PM	* Precipitation — Inches	Solar Energy on Horizontal Surface — Btu/Sq Ft/Day	Maximum	Time of Max.	Av. Condition	Velocity 3 Ft. from Window at Time N — MPH	Outside Air Temperature at Time N — °F	Glass Heat Flow — Btu/Sq Ft/Hour/°F — Based on P,Q, & 70°F	f_0 — Btu/Sq Ft/Hour/°F	Glass Heat Flow — Btu/Sq Ft/Hour/°F — Based on f_0 & 70°F	$\frac{Q}{M}$ ×100	$\frac{R}{M}$ ×100	$\frac{T}{M}$ ×100
1	Jan 18	SE	9	12	6	30	38	19	19			58.5	9:00A	46.0								
2	19	W	12	18	4	37	40	32	32	T		46.5	9:00A	39.6								
3	20	W	8	12	2	40	46	28	28		638	44.0	7:30A	33.8								
4	22	SE	16	22	13	47	52	41	40	.01	420	35.0	7:00A	29.3								
5	23	W	22	29	14	38	46	30	30	.01	606	49.3	1:30P	42.3								
6	24	S	8	10	3	40	47	24	24		636	44.3	9:30A	33.8								
7	25	W	16	22	11	42	51	33	33	.01	76	44.3	9:30P	35.5								
8	26	SW	8	12	5	36	41	29	29		674	45.0	7:30A	38.0								
9	27	SW	12	17	8	43	46	37	35	T	153	39.4	8:30A	32.5								
10	29	W	19	22	15	3	5	1	1	T	802	89.8	9:00A	86.5								
11	30	W	14	18	9	6	9	-1	-1	T	689	88.2	6:30A	74.0								
12	31	SW	15	20	10	28	35	12	9		716	69.8	6:00A	52.2								
13	Feb 1	SW	12	15	9	31	34	2	23	T	77	58.0	10:00P	46.8	6.0	23.5	49.8	3.16	48.2	80.7	85.9	83.1
14	2	N	9	11	5	19	22	15	15	T	215	65.0	6:30A	58.2	11.0	16.5	63.0			84.5	96.9	
15	3	S	7	11	4	20	27	8	8		393	69.2	6:30A	54.2	6.5	9.0	65.5	1.46	47.5	78.3	94.7	68.6
16	5	NE	10	12	5	27	29	21	21	T	587	57.2	10:30P	50.0	6.0	21.4	52.0	2.21	46.2	77.4	90.9	70.8
17	6	W	9	12	5	21	26	12	12		511	62.0	6:30A	56.2	6.0	14.0	59.5	1.60	45.5	90.6	96.0	73.4
18	7	NW	13	15	11	26	28	20	20	T	471	63.0	10:30P	53.5	5.0	19.8	52.0	2.46	49.2	84.9	82.5	78.1
19	8	N	13	15	10	6	12	-4	-4		778	89.5	10:00P	78.2	0.4	-1.6	61.0	2.47	70.0	87.4	68.2	78.2
20	9	NE	10	12	7	2	9	-11	-11		747	97.6	6:30A	79.2	7.8	-11.0	90.0	1.79	69.0	81.1	92.2	70.7
21	10	SE	8	12	4	24	34	1	-1		715	72.0	7:00A	51.5	2.0	1.0	63.8	1.73	59.0	71.5	88.6	81.9
22	12	SW	13	19	9	31	33	30	30	.05	153	50.0	10:30P	47.5	15.0	32.6	46.3	2.95	39.6	95.0	92.6	79.2
23	13	NW	17	22	8	14	20	10	10	T	629	72.3	2:00P	71.0	11.2	14.0	66.2	3.26	61.5	98.2	91.6	85.1
24	14	S	8	11	4	24	32	7	5		481	64.5	7:30A	51.6	3.0	8.5	59.3	1.92	55.0	80.0	91.9	85.3
25	15	NW	14	17	6	36	43	21	21		667	58.8	10:00P	42.0	1.4	21.0	43.5	1.65	40.5	71.4	74.0	68.9
26	17	SE	5	8	4	28	37	12	12		479	64.0	8:00A	43.7	1.0	15.0	48.0	2.07	50.4	72.8	80.0	84.0
27	19	NW	13	18	10	18	32	7	7	.08	200	73.0	10:00P	63.3	5.1	6.8	65.5	2.52	63.0	84.4	87.5	84.0
28	20	W	13	18	6	12	20	-2	-2		913	85.2	8:00A	70.5	3.0	0.3	67.2	1.88	60.5	82.7	78.9	71.0
29	21	SW	8	12	2	34	42	14	13		903	54.0	6:00A	40.2	0.6	15.5	46.5	1.99	49.0	71.8	83.0	87.5
30	23	W	19	26	12	14	18	9	9	T	640	76.6	3:30P	72.2	10.0	13.5	65.3	2.66	57.0	94.2	85.2	74.4
31	24	W	10	13	5	11	15	3	3	T	730	80.5	8:00A	68.5	2.8	8.2	59.0	1.90	54.0	85.1	73.3	67.1
32	26	NW	17	21	10	14	17	7	7	.06	384	72.2	4:30P	71.2	5.0	14.0	56.0	3.10	58.0	88.6	77.6	70.3
33	27	W	11	12	4	6	12	-3	-3		1015	86.4	8:00A	76.0	3.0	2.0	64.3	2.70	70.0	88.0	76.4	81.0
34	28	S	7	12	3	20	28	-1	-1		339	71.2	8:00A	54.3	1.8	3.5	60.4	2.17	61.5	74.3	84.8	86.4
35	Mar 1	S	7	11	3	34	39	17	17		371	51.0	7:30A	39.5	1.0	16.8	46.5	2.07	49.0	77.5	91.2	96.1
36	2	SE	10	16	5	45	51	29	29	.02	214	49.8	6:30A	29.1	1.0	28.8	36.0	1.35	30.5	58.4	72.3	61.2
37	5	SW	16	30	7	48	56	39	39	.24	711	45.6	10:30P	27.8	7.0	38.0	35.0	1.34	23.6	51.0	66.8	41.8
38	6	SW	21	30	12	35	39	29	29	.02	852	55.8	2:30P	45.7	14.0	31.0	47.8	1.76	33.0	71.9	75.7	59.1
39	7	W	9	11	8	35	37	31	31	T	441	46.2	10:30P	40.0	1.4	34.4	31.8	1.43	27.0	86.6	68.8	58.4
40	8	W	11	18	5	28	30	23	23	.04	203	56.5	6:30P	44.5	4.5	24.2	42.0	2.41	43.0	87.6	74.3	76.1
41	9	NW	14	18	8	26	29	20	20	T	1010	63.5	6:30A	54.3	5.0	21.0	51.0	2.45	51.8	85.5	80.3	81.6
42	10	N	14	18	9	22	28	15	15	.37	302	67.3	10:30P	54.2	9.0	19.5	57.2	1.97	45.0	88.0	85.0	66.9
43	12	S	13	17	9	28	33	14	14	T	1071	64.8	6:30A	51.3	19.0	13.8	72.2	3.56	63.5	73.5	103.4	91.0
44	13	SW	20	34	9	52	58	42	33		1175	36.0	6:30A	23.5	19.0	42.0	36.0	2.32	27.0	65.3	100.0	75.0
45	14	NW	15	18	7	28	34	23	23	.04	652	56.5	10:30A	52.2	10.0	28.0	44.0	2.90	44.2	92.4	86.7	78.2
46	15	SW	8	9	3	36	42	18	18		1005	52.0	6:30A	38.0	3.8	18.5	51.0			63.1	98.0	
47	16	W	14	18	2	49	54	40	39		1231	38.2	6:30A	23.9	2.0	40.8	27.2	1.70	24.5	51.8	61.2	54.1
48	17	SW	17	23	9	62	70	43	38		1229	30.6	7:30A	10.2	2.0	43.0	25.0	1.96	24.3	33.3	81.7	79.9
49	24	NW	8	12	5	32	37	25	25		316	50.4	9:00A	42.5	4.6	28.5	42.5	2.23	39.5	74.3	84.3	78.9
50	26	SE	11	19	5	48	59	37	33	.27	230	39.2	6:30A	24.0	1.0	37.4	28.4	2.41	31.6	56.3	72.5	80.6
51	27	NW	14	17	3	37	56	28	28	.26	169	52.8	8:00P	40.7	4.2	28.0	43.0	1.63	34.6	77.1	81.4	65.5
52	28	NW	9	10	4	34	42	22	22		1356	54.8	6:00A	41.3	3.5	26.0	43.3	3.21	48.0	75.4	79.0	87.6
53	29	N	4	7	2	50	61	29	29		445	42.7	6:00A	20.0	1.2	34.0	31.8	1.98	32.5	46.8	74.5	75.6
54	31	E	11	17	7	47	51	41	41	.11	385	34.2	8:00A	27.5	9.7	41.0	33.9	2.31	27.8	80.4	69.1	71.3
55	Apr 5	E	12	18	8	58	68	45	45		1508	29.8	7:30A	14.5	11.0	46.0	28.2	2.75	29.8	48.7	94.6	83.2
Av. All Tests			12	17	7	30	36	20	19		608	58.6		47.4	5.7	20.8	50.3	2.25	43.8	76.3	83.4	75.4

* T = Trace of Precipitation

[1]Wind Velocity Gradients Near a Surface and Their Effect on Film Conductance, by F. C. Houghten and Paul McDermott. A.S.H.V.E. Transactions, Vol. 37, p. 301.

[2]Surface Conductance as Affected by Air Velocity, Temperature and Character of Surface, by F. B. Rowley, A. B. Algren and J. L. Blackshaw. A.S.H.V.E. Transactions, Vol. 36, p. 429.

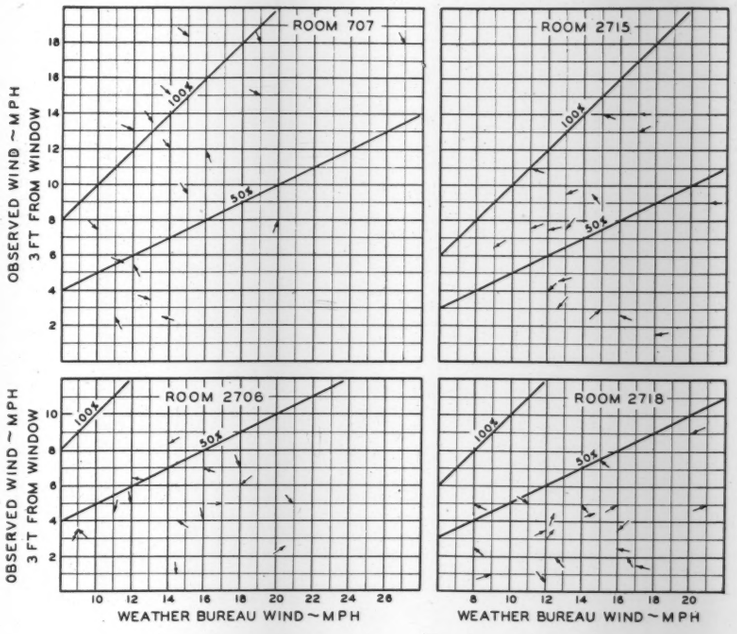


Fig. 4—Relation between observed wind 3 ft from window and wind recorded by Weather Bureau. All data for a northwest wind at Weather Bureau. Arrow points indicate magnitude and direction of wind to observer facing window from within. Lines indicate observed winds of 100 per cent and 50 per cent of Weather Bureau velocities

side air temperature in the center of the room and the glass surface temperature and the accepted inside film conductance coefficient, the discrepancy between this value and that calculated from the weather bureau is increased. The average for all tests shows the heat flow calculated from the observed temperatures and wind velocities outside of the window to be 83.6 per cent and the heat flow calculated from the inside film conductance coefficient to be 75.4 per cent of that based upon weather bureau observations. While these facts cannot be accepted as final proof of error in the practice of basing heat loss calculations directly on weather bureau data, they do represent strong evidence of such an error.

Considerable variation in the magnitude and direction of the wind velocity observed 3 ft outside of the window was apparent. It was also apparent that these velocities were usually lower than those recorded by the weather bureau. Fig. 4 shows the relation between the weather bureau wind velocities and those observed 3 ft outside of rooms 707, 2706, 2718 and 2715, for all observations in all tests when the weather bureau recorded a northwest wind. The values are given by the points of the arrows which indicate the direction in which the wind outside the window was blowing when observed by a person facing out of that particular window. For convenience in comparison, the lines, which represent respectively conditions where the wind velocity outside of the window was 100 per cent and 50 per cent of that recorded by the weather bureau, are drawn. This relationship is better expressed for the eight directions of wind recorded by the weather bureau and for the different windows listed, in Table 2, which gives the average and maximum percentages of all observed velocities outside of each of the seven windows in terms of the concurrent velocity recorded by the weather bureau. This table does not distinguish between direction of the velocities observed outside of the windows. The total number of observations on which the average is based is also given together with the number of observations for any particular window and wind velocity which exceeded 50 per cent of that recorded by the weather bureau.

Table 2 indicates a decided reduction in the observed wind velocities 3 ft from a window below that recorded by the weather bureau, and also a rather definite relationship between this reduction for any window and the direction of the wind. While in all cases the average wind velocity outside of the window is considerably less than that recorded at the weather bureau, certain windows, depending upon the direction of the wind, indicate occasional high velocities. This reduction in observed wind velocity near a building must necessarily be an important factor in reducing the actual heat loss below that estimated from weather bureau data. In this connection it should be pointed out that the velocity a few inches away from the glass or that actually affecting the surface heat transfer may be even a lower percentage of the weather bureau values, because windows are set in several inches from the plane of the outside of the building wall.

The relation between location of a room in a building and the average and maximum winds observed near it as given in Table 2 and Fig. 4 is of special interest.

Table 2—Wind Velocity 3 ft from Windows in Per Cent of Weather Bureau Velocity for Various Wind Directions

Wind Dir. / Room	North				Northwest				West				Southwest			
	Av. %	Max. %	No. Obs.	No. Obs. Over 50%	Av. %	Max. %	No. Obs.	No. Obs. Over 50%	Av. %	Max. %	No. Obs.	No. Obs. Over 50%	Av. %	Max. %	No. Obs.	No. Obs. Over 50%
705	23.5	39.0	8	0	27.1	60.0	20	1	35.4	88.0	12	3	36.5	95.0	10	2
707	34.9	79.0	9	3	70.8	122.6	17	11	70.6	105.0	6	4	68.4	117.0	12	9
1826	47.0	75.0	9	3	44.3	81.0	18	7	65.0	105.0	7	3	54.6	102.0	16	9
2718	17.0	23.0	7	0	28.5	62.5	19	1	52.2	98.0	10	3	36.5	100.0	15	2
2715	61.4	91.0	6	3	55.0	100.0	17	10	35.5	71.0	13	2	33.2	66.0	16	2
2703	32.7	50.0	7	0	38.9	73.0	22	4	23.9	52.0	10	1	55.4	106.0	17	9
2706	34.1	44.0	3	0	32.6	60.0	16	2	65.6	100.0	10	7	57.7	120.0	18	9
Dir. / Room	South				Southeast				East				Northeast			
705	38.4	62.0	8	1	70.3	91.0	6	5	90.0	90.0	1	1	90.2	114.0	5	4
707	38.5	70.0	11	3									46.0	55.0	3	1
1826	38.2	59.0	6	1	63.5	100.0	8	5	43.0	58.0	2	1	44.5	68.0	7	3
2718	50.1	77.0	7	2	55.0	92.0	9	5	41.0	58.0	3	1	27.4	39.0	7	0
2715	24.1	84.0	8	1	31.4	85.0	10	2	42.8	54.0	4	1	28.4	54.0	7	1
2703	31.7	96.0	9	2	20.5	52.0	10	1	51.0	75.0	6	4	53.0	68.0	5	3
2706	26.4	34.0	9	1	16.1	40.0	10	0	55.0	78.0	3	1	24.0	26.0	2	0

In general, the relation between wind velocity observed near the surface and the weather bureau wind is more or less logical. As an example, when the wind was from the northwest or when it was striking directly into the broad front of the building it tended to split sideways in each direction, giving high velocities for rooms 707 and 2715 which are near the corners of the building. Most surprising, however, is the fact that the highest velocities were experienced near certain 7th floor rooms where one would have expected shielding due to buildings across the street. As an example, the observed wind velocities were generally highest near room 707, having buildings

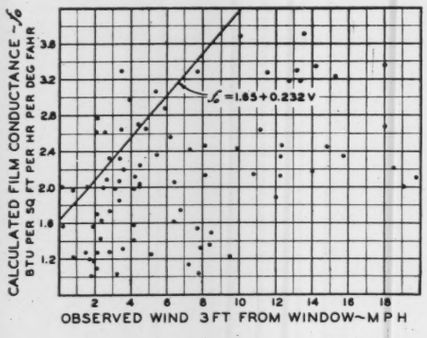


Fig. 5—Points showing relation between calculated f_o and observed wind velocity 3 ft from window. Curve giving values of f_o as determined by formula. Data from Room 707

extending to above this level on the opposite side of the street. Particularly was this true when the weather bureau reported a northwest, west or southwest wind, which seemed to enter Grant Street near Third Avenue and blow in a northeastwardly direction through the canyon formed by the taller buildings on both sides of Grant Street near and beyond Fourth Avenue. These facts do not agree with the frequent practice of assuming a lower wind for sheltered buildings. They seem to lend credence to claims concerning the howling winds through the canyons of New York and other large cities.

Fig. 5 shows the relation between the outside film conductance coefficient, when calculated by formula (3), for all observations on room 707 and the wind velocity observed outside the window. The curve giving values of $f_o = 1.65 + 0.232\ V$ is drawn for convenience in comparison. It will be observed that most of the values of f_o calculated from the inside air and glass surface temperatures are considerably lower than those given by the curve for the wind velocity observed 3 ft from the window. Similar data for all other rooms show approximately the same relationship, indicating the probability of a considerably lower actual rate of heat flow through windows than that calculated from wind velocities and temperatures observed 3 ft from the window.

The effect which wind has on the outside film conductance coefficient for a wall or window and therefore its effect on heat transfer is amply demonstrated by the work of the Research Laboratory[1] and Prof. Rowley[2] at the University of Minnesota. These laboratory studies, however, gave results for definitely controlled conditions where the wind was parallel to the wall surface. This does not necessarily apply for natural wind around a building.

The effect which wind has on heat loss from a building differs widely according to variations in wall construction and the percentage of wall and window areas in the building. The effect of wind may be negligible for a well insulated wall, while for a single thickness of glass it may become a very important factor.

The exposed walls of the eight rooms studied in the Grant Building contained 42 per cent of glass and metal sash and frame area, and 58 per cent of masonry walls. The masonry walls had a conductance of 0.413. Assuming an inside film conductance coefficient of 1.65 and an outside brick film conductance coefficient of $f_o = 2.0 + 0.4\ V$, where V is the wind velocity, the heat transmission coefficient for the wall becomes 0.283 and 0.317 Btu per square foot per hour per degree temperature difference for wind velocities of zero and 15 mph. Assuming glass surface film conductance coefficients $f_i = 1.65$ and $f_o = 1.65 + 0.232\ V$, and allowing no resistance to the flow of heat for the glass itself, the heat transfer coefficient for the glass becomes 0.825, and 1.248 Btu for the same wind velocities. The 15 mph wind increases the heat transfer through the glass to 151 per cent of that for still air, which becomes a considerable percentage increase for the total building.

Summary

This report, dealing with wind velocities and temperatures near the windows of several rooms, and their relation to weather bureau observations and heat loss through windows, represents an analysis of a small phase of the study made in the Grant Building during the past heating season. As such it should be considered as a single step in the analysis of a much more comprehensive study. The results suggest reasons for shortcomings in the present methods of estimating heating requirements of buildings based upon improper application of local weather bureau records to conditions in the immediate environment of a building. The findings here presented should not be considered final and conclusive in demonstrating possible errors in the present methods of estimating heat losses. They should be accepted rather as strong evidence of such errors to be considered in relation to other evidence resulting from more complete analysis of the data already collected in the Grant Building and elsewhere and to future studies, which will probably be made in other types of buildings and in other localities.

Acknowledgment

The authors wish to acknowledge the valuable assistance rendered the Research Laboratory by the Grant Building Management in making the rooms and other facilities available and in otherwise helping to make the study a success. Acknowledgment is also due to Prof. John A. Dent and the Mechanical Engineering Department of the University of Pittsburgh for cooperating in the analysis of the data through the work of student assistants, and for the assistance of Mr. Fred Hogue, a graduate student of the University of Pittsburgh, who aided in the investigation and who is preparing a thesis on the subject for the degree of Master of Science.

What Is the Cooling Load Factor in Air Conditioning?

By **John Everetts, Jr.,**† **New York, N. Y.**

(MEMBER)

AIR conditioning, especially summer cooling, is becoming more a necessity than a novelty, particularly in department stores, office buildings, restaurants and other places of business where cooling for comfort is a primary objective.

One of the most important considerations now being given to systems of this type is the operating cost. The forecasting of this operating cost with any degree of accuracy is a complex problem, because it is dependent largely upon the cooling load factor. This factor determines the operating hours of the refrigerating plant required in such an air conditioning system and as this is the most expensive part of the system to operate, it is important that this factor be properly calculated. The elements involved in making up the cooling load factor are as follows:

1—Heat transmission through building
2—Sun effect
3—Internal light and power load
4—People load
5—Outside air load

The variables that affect the cooling load factors are geographical location and type of installation. The geographical location determines the outside design condition and the portion of the cooling load factor affected by the outside dry bulb and wet bulb temperatures. These outside design conditions for different cities are given in Table 1, Chapt 8, p. 117. (See The A. S. H. V. E. Guide, 1934.) Knowing these conditions and the maintained conditions required, the heat transmission, sun effect and outside air load may be calculated in the recommended manner. It must be borne in mind, however, that the load calculations made under these conditions govern the size of the equipment required to maintain the guaranteed temperature and humidity within the conditioned space at all times. These design temperatures shown in THE GUIDE are not maximum temperatures, nor average temperatures, but they have been chosen after considerable study of the Weather Bureau records as the temperature that is not exceeded more than 5 to 8 per cent of the time during June, July, August and September for an average year. Therefore, the refrigeration load made up by the transmission, sun effect and outside air is a maximum only 5 to 8 per cent of the season and ranges from zero to maximum 92 to 95 per cent of the time. As these are such variable quantities, it is necessary to determine the average load for the season in order to establish a basis for calculating the operating costs.

Table 1 shows the number of hours during June, July, August and September that different dry bulb and wet bulb temperatures occur. This table has been worked out from the Weather Bureau records of the Central Park Station, in New York, and is an average of hourly

Table 1—Summation for Years 1925-1931 Inclusive of Hourly Dry Bulb and Wet Bulb Readings June-Sept. Inclusive. Central Park, New York City

Data based on 24 hour readings.

TOTAL HOURS AT WET BULB TEMPERATURES GIVEN.		TOTAL HOURS AT DRY BULB TEMPERATURES GIVEN.	
*WET BULB TEMP. F	AVERAGE	DRY BULB TEMP. F	AVERAGE
81	2	101	1
80	4	100	1
79	7	99	1
78	16	98	2
77	22	97	4
76	37	96	5
75	42	95	4
74	62	94	6
73	70	93	8
72	121	92	13
71	119	91	15
70	165	90	16
65-69	778	89	18
60-64	710	88	29
55-59	458	87	34
50-54	221	86	49
45-49	72	85	49
40-44	22	84	61
		83	62
	2,928	82	72
		81	72
		80	95
		79	81
		78	106
		77	104
		76	125
		75	134
		74	147
		73	159
		72	160
		71	144
		70	158
		69	125
		68	133
		67	106
		66	104
		65	87
		64	79
		63	71
		62	58
		61	45
		60	40
		59	35
		58	29
		57	23
		56	18
		55	14
		54	9
		53	5
		52	4
		51	2
		50	2
		49	1
		48	1
		47	1
		46	1
		45	..
			2,928

*Wet bulb data furnished through courtesy of J. C. Albright, N. Y.

†Engr., W. L. Fleisher, Consulting Engr.

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Buck Hill Falls, Pa., June, 1934.

readings taken over a period of the last seven years. Table 1 gives the data for 24-hour operation. The type of system, which may be classified as either industrial process conditioning or comfort cooling, determines which set of tables to use; that is, an industrial process conditioning system operating 24 hours a day to maintain given conditions, would have its cooling load factor based on the data given in Table 1. A comfort cooling installation, such as an office building or a department store, would use a similar table showing data from 8 a. m. to 5 p. m., whereas a theater cooling system would use a table showing the data from 11 a. m. to 11 p. m.

To show the practical application of these tables in determining the cooling load factor, two examples are given: first, an industrial plant operating 24 hours a day, and the other a department store operating from 8 a. m. to 5 p. m. The design data for the industrial plant is tabulated as follows:

1—Industrial Plant, New York City

a—Outside design conditions.....	95 D. B.		75 W. B.
b—Inside maintained conditions..	68 D. B.		57 W. B.
c—Heat Transmission (no sun effect)	270,000	Btu/hr	45.5 per cent
d—Electric power and light.....	70,600	"	11.8 per cent
e—People	45,900	"	7.7 per cent
f—Outside air	208,000	"	35.0 per cent
g— TOTAL LOAD	594,500		100 per cent

h—Tons refrigeration required—49.6 tons.

To determine the cooling load factor, it is necessary to calculate the total number of ton hours per season that this plant would require. Table 2 shows the tabulation of the temperature, hours, and ton hours per season for the heat transmission through the building. These calculations have been made for each temperature from 101 F down to the inside maintained temperature at which condition the heat transmission would be zero. The sum total of the ton hours per season for the heat transmission in this case is 15,167.1, as shown.

Table 2—Variation of Load from Heat Transmission (Industrial Plant)

Dry Bulb Temp.	Hours	Ton Hours per Season	Dry Bulb Temp.	Hours	Ton Hours per Season
100	1	26.7	86	49	734.0
99	1	25.8	85	49	694.0
98	2	50.0	84	61	812.0
97	4	96.7	83	62	774.0
96	5	116.5	82	72	840.0
95	4	90.0	81	72	780.0
94	6	130.0	80	95	950.0
93	8	166.7	79	81	742.0
92	13	261.5	78	106	883.0
91	15	287.5	77	104	780.0
90	16	293.0	76	125	833.0
89	18	314.5	75	134	780.0
88	29	483.0	74	147	833.0
87	34	538.0	73	159	782.0
			72	160	534.0
			71	144	359.0
			70	158	263.5
			69	125	104.2
				2,060 hr	15,167.1

The heat loads from the lights, motors and people remains constant throughout the season but as they are only effective a total of nine hours daily, the total ton hours per season would be:

$$\text{Electric } \frac{70{,}600}{12{,}000} \times 2{,}928 \text{ hours/season} \times \frac{9}{24} = 6{,}460 \text{ ton hours/season}$$

$$\text{People } \frac{45{,}900}{12{,}000} \times 2{,}928 \text{ hours/season} \times \frac{9}{24} = 4{,}210 \text{ ton hours/season}$$

The outside air load is calculated in the same manner as the heat transmission and is tabulated as shown in Table 3, the difference, however, being that the outside air load from which the total heat must be removed is based on the wet bulb temperature and not the dry bulb. The sum total ton hours per season in this case is 19,915.8, as shown.

Table 3—Variation of Load from Outside Air (Industrial Plant)

Wet Bulb Temp.	Hours	Ton Hours per Season
81	2	48.6
80	4	94.0
79	7	155.2
78	16	346.0
77	22	435.0
76	37	685.0
75	42	727.0
74	62	1,004.0
73	70	1,057.0
72	121	1,700.0
71	119	1,538.0
70	165	1,958.0
65-69	778	6,900.0
60-64	710	3,083.0
57-59	229	185.0
	2,384	19,915.8

The total ton hours per season and their percentages of the total are as follows:

a—Transmission	15,167.1 ton hours/season	33.1 per cent
b—Electric	6,460.0 ton hours/season	14.1 per cent
c—People	4,210.0 ton hours/season	9.2 per cent
d—Outside air	19,915.8 ton hours/season	43.6 per cent
e—Total	45,752.9	100.0 per cent

For a 24-hour operation, there are a total of 2,928 hours per cooling season; therefore, the average load would be $\frac{45{,}752.9}{2928} = 15.6$ tons. The refrigeration required to operate the plant at the design conditions was shown to be 49.6 tons; therefore, the load factor would be $\frac{15.6}{49.6} \times 100 = 31.5$ per cent.

Knowing the operating cost of the refrigerating system per ton of refrigeration per hour at full load; it is merely a matter of simple arithmetic to multiply this cost by the total ton hours per season to arrive at a seasonal operating cost, or if the full load operating cost per hour is known, this may be multiplied by the 2,928 hours per season and the load factor of 31.5 per cent to give the same results.

The second example showing the determination of the cooling load factor for a department store, based on 8 a. m. to 5 p. m. operation, may be calculated in the same general manner as shown for the industrial plant. The data for this particular installation is given below:

Outside design conditions.....	95 D. B.	75 W. B.
Maintained conditions	80 D. B.	67 W. B.

Heat Load in Btu/hour and Per Cent of Total

	Basement	Per Cent	1st Floor	Per Cent	Total	Per Cent
Transmission ...	45,000	6	286,000	17½	331,000	13½
Sun Effect	15,000	2	19,000	1	34,000	1½
People	300,000	38	640,000	38	940,000	38¾
Lights	200,000	25½	222,000	13½	442,000	17¼
Outside Air	225,000	28½	480,000	30	705,000	29
	785,000	100	1,647,000	100	2,432,000	100
Tons Ref.	65		137		202	

Table 4 shows tabulation of the ton hours per season for the transmission load, giving a sum total for the basement of 593.9 and for the first floor 3,737, making a total of 4,330.9, as shown.

Table 4—Variation in Load from Heat Transmission (Dept. Store)

Dry Bulb Temp.	Hours	Basement Ton Hours per Season	1st Floor Ton Hours per Season
101	1	5.25	33.10
100	1	5.00	31.80
99	1	4.75	30.20
98	2	9.00	57.00
97	4	17.00	108.00
96	5	20.00	127.00
95	4	15.00	95.00
94	5	17.50	111.00
93	7	22.70	145.00
92	11	33.00	210.00
91	14	38.40	245.00
90	14	35.00	222.00
89	15	34.70	214.00
88	23	46.00	292.00
87	29	50.70	323.00
86	42	63.00	400.00
85	40	50.00	318.00
84	46	46.00	292.00
83	47	35.20	224.00
82	54	27.00	171.50
81	55	13.70	87.40
80	65	0	0
	486 hours	593.90	3737.00

The total ton hours per season for sun effect is calculated on the basis that the sun shines only 70 per cent of the 12 possible sunlight hours per day during the cooling season. As the store operation is only nine hours per day, 75 per cent of this value is taken. Calculated in this manner, the total ton hours per season for sun effect in the basement would be 960 and for the first floor 1,218, making a total of 2,178 ton hours per season.

The electric load in a department store would be continuous for the 1,220 hours per season; therefore, the total ton hours per season for the basement would be 20,350 and for the first floor 21,600, making a total of 42,950.

The people load in a department store is quite variable, however, and an average of 60 per cent of the full load capacity may be given as a fair figure. The ton hours per season for the people in the basement on this basis would be 18,300 and in the first floor 39,100, making a total of 57,400.

Table 5 shows the ton hours per season at the different wet bulb temperatures, the sum total being for the basement 5,780 and for the first floor 12,335, making a total of 18,115. The tabulated ton hours per season and their percentages of the total are as shown below:

Table 5—Variation in Load from Outside Air (Dept. Store)

Wet Bulb Temp.	Hours	Basement Ton Hours per Season	1st Floor Ton Hours per Season
81	2	72	150
80	4	129	276
79	6	177	377
78	14	373	798
77	19	456	973
76	31	662	1412
75	30	563	1200
74	45	726	1558
73	42	577	1232
72	67	758	1618
71	49	440	935
70	74	490	1050
69–67	165	357	756
	548	5780	12335
			5780
			18115

	Basement	Per Cent	1st Floor	Per Cent	Total	Per Cent
Transmission	594	1.30	3,737	4.75	4,331	3.5
Sun Effect	960	2.00	1,218	1.25	2,178	1.5
People	18,300	40.00	39,100	49.50	57,400	46.0
Lights	20,350	44.20	22,600	28.75	42,950	34.5
Outside Air	5,780	12.50	12,335	15.75	18,115	14.5
	45,984	100.00	78,990	100.00	124,974	100.0

$$\text{Tons Refrigeration} \quad \frac{45{,}984}{1{,}220} = 37.7 \qquad \frac{78{,}990}{1{,}220} = 64.7 \qquad \frac{124{,}974}{1{,}220} = 102.4 \text{ Tons}$$

The refrigeration required as previously shown, to maintain the proper conditions within the store, was 202 tons; therefore, the cooling load factor would be

$$\frac{102.4}{202} \times 100 = 50.5 \text{ per cent.}$$

The operating cost may be calculated in the same manner as previously mentioned.

Summary—Conclusions

1. As the cooling load factor is made up of many variables, it is impractical to use any one load factor as a criterion for other similar installations unless the items making up the load are approximately the same in proportion.
2. The load factor will be different for similar installations located in different parts of the country because of the difference in the design conditions.
3. The cooling load factor is only to be used to determine the operating cost of the refrigerating end of the system and does not in any way affect the operating cost of other equipment, such as pumps and fans which are generally operated at full load throughout the season.

A. S. A. Advisory Committee

An announcement by Howard Coonley, president of the *American Standards Association,* tells of the membership of nine of the nation's industrial leaders on the A. S. A. Advisory Committee.

G. B. Cortelyou, president, Consolidated Gas Co., who has served as Postmaster General and later as Secretary of the Treasury, will be chairman of the Advisory Committee, whose members are: S. L. Avery, chairman of the board, Montgomery Ward & Co., president, U. S. Gypsum Co.; Lammot DuPont, president, E. I. DuPont de Nemours & Co., chairman, General Motors Corp.; W. B. Gifford, president, American Telephone & Telegraph Co.; H. I. Harriman, president, Chamber of Commerce of U. S. A.; W. A. Irvin, president, U. S. Steel Corp.; J. H. McGraw, chairman of the board, McGraw-Hill Publishing Co.; Gerard Swope, president, General Electric Co.; and Daniel Willard, president, Baltimore & Ohio Railroad.

More than 250 standards and codes have been developed by the Association and nearly 200 are under development or are being revised by the 3,000 engineers, scientists, and industrialists, who serve on its committees.

A Method of Testing Filters Against Pollens, Bacteria and Molds*

By **Philip Drinker**[1] **and Wm. F. Wells**[2]

Boston, Mass.

IN all methods for the testing of filters the dispersion of the test dusts in air and the maintenance of dust suspensions of constant composition is a difficult technic. It is well known that the more dilute the suspension the easier it is to maintain in stable form, but this limitation in turn restricts the methods of measuring concentration to those of unusual sensitivity. Low concentrations obviously are unsuited to the determination of dust-holding capacity. In spite of these objections, it is believed simple sensitive methods which can be duplicated by anyone accustomed to handling standard laboratory equipment may be of interest.

In the present paper, the application of pollen, bacteria, and molds to commercial filter testing for which purpose these substances possess some very practical advantages, will be described. In brief, the method consists in suspending pollen, bacteria, or molds in air and drawing this air through filters or a section thereof at whatever rate the filter manufacturer specifies. Two Wells' air centrifuges[3] sample the cleaned and uncleaned air simultaneously and the pollen or bacteria counts on the two samples are then compared.

Up to the present time filter testing against pollen or bacteria has been carried out by the settlement method. Petri dishes, suitably coated, are exposed to affluent and effluent sides of the filter and the number of pollen grains or bacterial colonies are counted on each dish. As a variation of this technic, microscope slides have been substituted for the dishes. In either case reliance is placed upon the random or chance settlement of the pollen or bacteria. As the velocity of the air stream entering or leaving a commercial filter will greatly exceed the critical transporting velocity for common bacteria, and even common pollen, it is difficult to find any excuse except convenience to recommend these methods.

Pollen.—In Fig. 1 is shown under about 70 diameter magnification, pollen of the giant ragweed type. The grains appear somewhat spherical with exfoliations, the general effect being roundish. Two pair of pollen grains are touching at one point while another grain shows some fan-like impurity sticking to the edge. Obviously much of the material shown is not pollen but miscellaneous matter found in air anywhere. The purpose of showing the photograph is to illustrate how easy it is to identify pollen in the presence of these impurities. Twenty-one grains appear in this photograph which is of exactly the magnification we use in the testing procedure. If one has trouble identifying the pollen it can be stained by various methods, but slight practice, in our experience, is sufficient to let one dispense with stains.

The pollen used and illustrated here is obtained from Knapp & Knapp, North Hollywood, California, at a cost of about $50.00 per kg. This price sounds ominously high but it must be remembered that nature inflicts upon us pollen concentrations rarely exceeding a few grains per cubic foot, while 1 g of pollen contains about 3×10^8 individual grains. Gross wastage in making suspensions is inevitable but a kilogram of pollen will last for many tests.

Fig. 1—Pollen grains caught by the Wells centrifuge on thin celluloid foil and photographed at 70 diameters magnification. Twenty-one distinct grains appear with a possible 22nd touching some extraneous matter near the edge at 11 o'clock. Errors of this magnitude are inherent in the method

In Fig. 2 are shown under 70 diameters magnification, household dust particles collected by exposing a slide for a time on the top shelf of a book case, about 8 ft above the floor. In this photomicrograph are many particles both larger and smaller than pollen but it is obvious that a filter which does not catch pollen will fail to catch a large proportion of miscellaneous household dust.

Bacteria and molds.—In the case of bacteria and molds, colonies and not individual organisms are counted, macroscopically. When the number of colonies on a tube

From the Departments of Industrial Hygiene[1] and Public Health Administration[2], Harvard School of Public Health, Boston, Mass.

*Presented at March, 1934, meeting of the Massachusetts Chapter at the Harvard School of Public Health.

[3]Apparatus for Study of the Bacterial Behavior of Air, W. F. Wells, *Am. Jour. Pub. Health,* Jan., 1933, vol. 23, p. 58.

is very great the counting may be easier microscopically than macroscopically and it is then preferred to count fields under low magnifications of ten to fifty with a binocular dissecting microscope although the ordinary microscope with a 32 mm objective can be used.

B. coli are readily obtainable in pure strains everywhere and can be grown on special selective media which practically eliminate contaminating bacteria likely to occur in the test air. These bacteria are quite harmless to man when breathed as was demonstrated when enormous numbers from street manure were blown about in the days of horse-drawn vehicles.

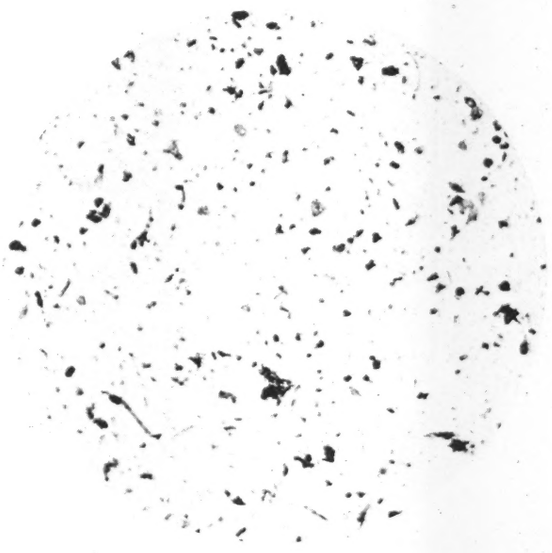


Fig. 2—Dust deposited on an uncoated microscope slide placed in a large library reading room. Magnification 70 x

In Fig. 3*A* and *B* are shown macroscopic photographs of *B. coli* colonies as sampled by the Wells apparatus and incubated for 24 hours at 37 C. These colonies can be counted either macroscopically or microscopically according to personal preference. Micro counting gives wider range of test concentrations and so reduces the danger of getting indeterminate test results after incubation.

The convenience of using mold spores was discovered quite by accident. A culture of molds growing on the wick of a wet bulb thermometer in the air line was found to be passing a continuous stream of mold spores which were readily determined on incubation. This source of contamination thus became a convenient means for testing the penetrability of filters to molds.

Testing procedure.—In the work reported only filter substances which are of the cloth or paper type and adapted to small scale tests have been used. Makers of certain of the viscous or impingement type filters claim that small sections of their filters are not representative of the full sized unit. That issue or objection was avoided by testing only the cloth and paper types which were selected simply to make possible a study of the method—not to derive performance data on any one make of filter.

Suitable sections are cut from the filters and tied with fine thread around small wire cylinders through one end of which is pushed a glass tube in a suitable rubber stopper. A similar glass tube is placed close to the filter under test. Both tubes lead to Wells centrifuges equipped with flowmeters and clamps for controlling airflow. Air velocities through the filter substances are adjusted to conform to rates specified by the makers of the filter materials.

Pollen suspensions are maintained in a glass-lined tank (140 cu ft) by blowing pollen in through the special Tee-tube arrangement with the two flowmeters and electric bell shown in the center of Fig. 4. Bacteria are blown in by a simple aspirator of the surgical nasal spray type. Within the tank an oscillating desk fan keeps the air continually in motion.

In pollen testing the tube of the Wells machine (Fig. 5*A*) is lined with a small piece of thin celluloid foil (Fig. 5*B*) on which the pollen is thrown by centrifugal action. At the end of the run the foil is removed, a vertical strip about 5/8 in. wide is cut with scissors and fastened, pollen side up, on a clean microscope slide with amyl acetate. With a mechanical stage on the microscope several traverses are made the full length of the slide and all the particles coming within a definite area in the eye piece of the microscope are counted. The average count is recorded and if the variation is too great further mounts are made and checked.

Fig. 3*A*—*B. coli* caught by the Wells machine on the affluent side of a test filter

Fig. 3*B*—Same on effluent side. Both photographs are full size. Note that colonies are larger on the cleaned air side

For bacterial suspensions a 24 hour broth culture was diluted 10 times and 25 cc of this diluted mixture was atomized into the tank. The rapid evaporation of the droplets leave bacteria suspended in the tank air. By using different dilutions, virtually any desired bacterial concentration can be obtained.

In the case of molds it is only necessary to allow normal air to pass through a tube containing the culture and molds are picked up and diffused in the tank. The concentrations resulting are easily controlled.

In all sampling it is advisable to adjust airflows through the two intakes of both sampling tubes to insure comparable rates, and to run each for time intervals such that the deposits on the collecting tubes will be nearly alike. The most suitable air volumes and velocities can only be obtained by practical trial.

Discussion.—Comparing pollen, bacteria, and molds with the usual dusts, fumes, or smokes used extensively in the laboratory for testing filters, it is estimated that the plant or wind-borne pollen is relatively easy to catch, owing mainly to its large size (17 microns), while the bacteria and molds are about as hard to catch as well-dispersed lead fume but not so difficult as tobacco smoke.

The advisability or ethics of rating filters for bacteria or pollen in terms of per cent efficiency is seriously questioned. The dust loading and resistance problems in these cases simply do not exist. The purchaser, and probably the manufacturer has no idea what normal pollen or bacterial count in air may be. Pollen filters should be proof against passage of pollen when airflow is at the rate specified by the manufacturer. The customer should demand and be able to receive filters which are 100 per cent efficient against air-borne pollen. Whether he is to receive filters which are effective against bacteria will depend entirely on the demand. It is believed that there will be a definite field for bacterial filters for use in connection with hospitals. At present no commercial filters which attain high bacterial filtering efficiency in terms of the test methods described are known.

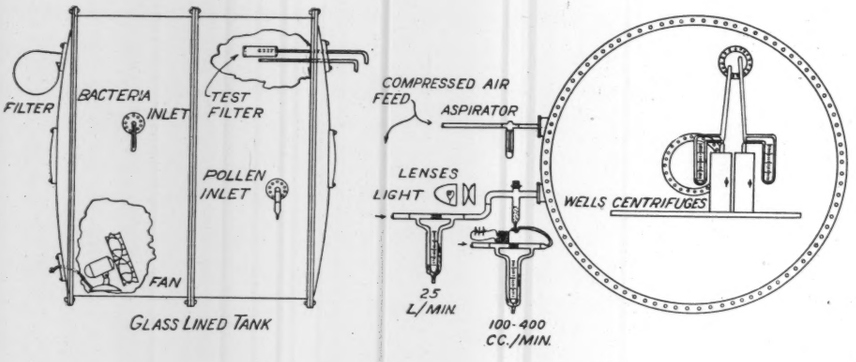


Fig. 4—Test set-up for pollen and bacterial filters. Filter at upper left is vent to replace air withdrawn by the sampler

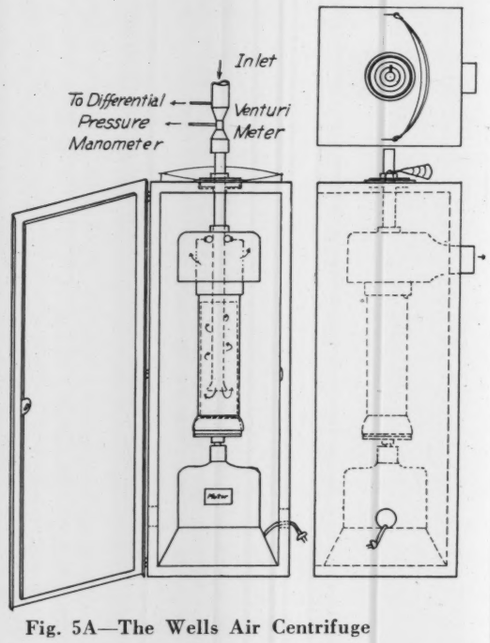


Fig. 5A—The Wells Air Centrifuge

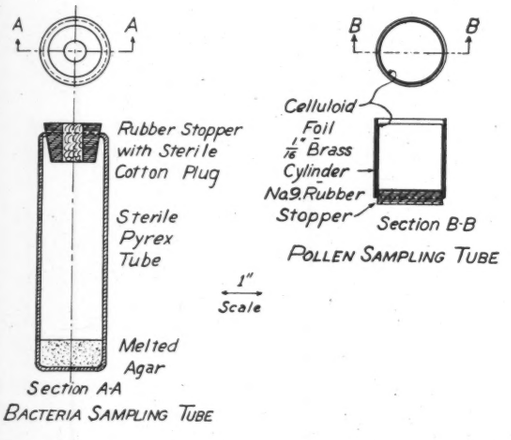


Fig. 5B—Bacterial and pollen sampling tubes for Wells centrifuge

Summary

A method of testing commercial filters for cleaning the air of buildings is described. The test suspensions consist of (1) air-borne pollen of the giant ragweed type with particle size of 17 microns, (2) harmless and easily identifiable bacteria such as *B. coli,* and (3) a common type of mold. Suspensions are set up in a tank of suitable size and maintained by means of an oscillating fan with the addition of further material as needed. By means of Wells air centrifuges, two samples of air from the tank are taken simultaneously, one passing through a section of the filter under test and the other coming direct from the tank. Comparative counts of the pollen grains or of the bacterial colonies are then made and performance judged from the results.

Coming Meetings

October 1-5, 1934—*American Welding Society,* Hotel New Yorker, New York City.

December 27-30, 1934—*American Association for the Advancement of Science,* Mellon Institute, Pittsburgh, Pa.

January 10-12, 1935—*American Engineering Council,* Washington, D. C.

Buffalo Prepares for 1935

The Society's Annual Meeting in January, 1935, will be held in Buffalo, New York, at the Hotel Statler and the members of the Western New York Chapter will be hosts, according to the announcement of the Council.

Two meetings of the Western New York officers and members have been held to discuss plans and committee organization. The latest get-to-gether was on August 16 when Capt. Park Hedley piloted a group for nine holes of golf before boarding his motor cruiser for a trip on Lake Ontario. A happy group of Western New York Chapter officers and past presidents is shown on the pier of the Olcott Yacht Club.

Admiral Reeves Addresses Engineers at New York

August 8, 1934. A successful pre-season meeting was held by the New York Chapter of the Society. The feature of the meeting was an address by Admiral Joseph M. Reeves, Commander in Chief of the U. S. Naval Fleet, who addressed a joint meeting of eight engineering societies, among which were the *Society of Naval Architects and Marine Engineers, American Institute of Electrical Engineers, American Institute of Mining & Metallurgical Engineers, American Society of Civil Engineers, American Society of Mechanical Engineers,* the *Society of American Military Engineers,* and the *Society of Terminal Engineers* at the Engineering Societies Bldg.

Prior to Admiral Reeves address, moving pictures showing fleet manouvers were enjoyed.

Admiral Reeves, who is the highest ranking officer of the U. S. Navy afloat, spoke on The U. S. Navy, and said that the present high state reached by this country's navy was only possible through the cooperation of engineers. He classified the different types of naval forces according to their use, and told the purpose of each in naval strategy.

Admiral Reeves was an officer in the engineer force of the Oregon in the famous cruise around Cape Horn. He was awarded the Naval Cross for exceptional meritorious service in the World War. He has been a member of the staff and head of the Tactics Department in the Naval War College, and later commanded and organized the Air Force of the battle fleet.

The reception and dinner tendered Admiral Reeves by the officers of the 8 participating societies was held at 6:30 p. m. in the Engineers Club. C. V. Haynes President of the Society and H. W. Fuller, President of the New York Chapter, represented the A. S. H. V. E.

Prior to the joint meeting, members of the New York Chapter of the Society dined at 14 East 60th St.

Consulting Offices Opened in New York and Hempstead

New offices have recently been opened by H. V. Cornwell and E. C. Blackburn, Jr., consulting engineers, at 250 Fulton Ave., Hempstead, L. I., N. Y., and 570 Lexington Ave., New York City. Announcement has been made of their plans to practice engineering, specializing in the design of power plants and heating, plumbing, electrical wiring, ventilating, air conditioning, refrigeration and sprinkler systems for buildings of all sizes and types. Mr. Blackburn is a licensed professional engineer in the states of New York and Oregon and his experience covers service with consulting engineers and architects in both the east and west.

Echos of Buck Hill

The pictures and proceedings of the Buck Hill Falls meeting of the Society, which appeared in the August Journal, recalled to many members, the enjoyable time spent in the Poconos during June. The splendid scenic views were photographed by J. J. Hayes, Secretary of Illinois Chapter, who did some extensive exploring with his camera.

The Society's leading golfer, J. F. McIntire, winner of the Research Cup and other tournament trophies at Buck Hill Falls, was persuaded to pose with the Cup and the faithful "One Putt Calamity Jane".

Cleveland

May 24, 1934. The meeting was called to order by Pres. F. A. Kitchen at 8:00 p. m. in the meeting room of the *Cleveland Engineering Society.* Thirty-eight members and guests were present.

The minutes of the April meeting were read and approved and due to the absence of H. M. Nobis the annual report of the treasurer was deferred until the June meeting. President Kitchen then called for reports from the Membership Committees.

Philip Cohen, chairman of the Red Team, reported that he had received the application of K. W. Schick for membership in the Society. He then nominated Mr. Schick for membership in the Cleveland Chapter, election to become effective at the time of his election to the A.S.H.V.E. The nomination was seconded and carried.

President Kitchen spoke briefly about the Semi-Annual Meeting of the Society and asked for the raising of hands of all members expecting to attend.

The Nominating Committee was then asked to report. As the committee was not prepared at this time, it was decided that their report would be mailed to the members of the Chapter together with a letter ballot. President Kitchen emphasized the fact that the members could substitute the names of any other members in good standing for those submitted by the Nominating Committee. Following this business the speakers of the evening were introduced.

The first speaker, L. T. Avery, discussed the subject, Mechanical Refrigeration Applied to Air Conditioning. Mr. Avery reviewed briefly the fundamentals of air conditioning explaining what the term meant and what the engineer tries to accomplish. Mention was made of the comfort chart and how refrigeration could be applied to obtain summer comfort conditions. He pointed out that cooling could be accomplished by the use of ice, refrigeration applied to a separate cooler, refrigeration used in connection with an air washer, and refrigeration used with direct cooling coils. A numerical example illustrated the fact that the cheapest method of producing a given amount of cooling is accomplished by means of direct cooling coils but the speaker stated that other conditions might make it advisable to use one of the other methods.

Following Mr. Avery's talk W. E. Stark spoke on Air Conditioning with Solid Adsorbents. Mr. Stark mentioned the various materials that could be used as solid adsorbents but stated that silica gel and activated alumina were among the

most important. By means of equilibrium curves he explained how the material could extract water vapor from the atmosphere and how the latent heat plus the heat of adsorption increased the temperature of the air in a silica gel plant illustrating his talk by means of line diagrams. After explaining how the dehydrator was constructed, he gave comparative figures of operating costs on installations that were sponsored by the *American Gas Association*.

The final talk, Steam Jet Refrigeration for Air Conditioning, was presented by F. R. Dickenson. Mr. Dickenson stated that steam jet refrigeration was not a new idea but had been patented by Parsons late in the nineteenth century. He discussed the fundamentals of producing a vacuum by means of a high velocity jet and, with the aid of a slide showing diagrammatically a steam jet apparatus, he explained how the machine functioned. Several other slides were shown to illustrate the range of sizes in which these machines are made.

Considerable discussion of these three papers were entered into by the members and guests. President Kitchen expressed the thanks of the Chapter to the speakers and the meeting adjourned at 10:30 p. m.

Kansas City

May 10, 1934. The Kansas City Chapter met on May 10, 1934, 6:00 p. m., at the Newbern Hotel, with an attendance of 26 members and 9 guests.

After dinner was served the meeting was called to order by David Caleb, president.

L. R. Chase, Chairman of the Committee on Social Activities, in charge of arrangements for the summer outing, reported that his committee wished to recommend Wildwood Lakes to the Chapter as a desirable place for the June outing. Mr. Chase was instructed, on vote by the Chapter, to arrange for the outing to be held at Wildwood Lakes on some Tuesday night in June.

L. A. Stephenson, Chairman of the Air Conditioning Committee which was appointed in February to assist other members of the industry in sponsoring a series of educational lectures on the subject of air conditioning, reported for his committee. This committee recommended that these lecture programs be discontinued for the present, chiefly for the reason that the attendance had not been what it should have been, and recommended further that they be resumed next year if interest seemed at that time to justify doing so. The President expressed his thanks to the committee for their hard work in this connection.

E. K. Campbell, Chairman of the Constitution and By-laws Committee, presented a revised draft of the Constitution of the Chapter. After an extensive discussion on the same it was unanimously adopted on motion by Mr. Stephenson and seconded by J. G. Lewis. The Secretary was instructed to file a copy with the records of the Chapter and to send a copy to A. V. Hutchinson, national secretary of the Society.

The President appointed an Auditing Committee consisting of Messrs. Carl Clegg and L. R. Chase, to audit the books of the treasurer and transfer his accounts to the new treasurer to be elected at the meeting.

The Nominating Committee, consisting of Messrs. Campbell, Russell and Clegg, made the following nominations for officers and board of governors for the ensuing year:

President—L. A. Stephenson.

Vice-President—C. A. Weiss.

Secretary—L. R. Chase.

Treasurer—G. L. Bliss.

Board of Governors—David Caleb, C. A. Flarsheim, and Harry Filkins.

Upon motion by N. W. Downes seconded by J. H. Kitchen, the secretary was instructed to cast the unanimous ballot of the Chapter for the above named men, declaring them elected.

Referring to the death of Dean Anderson, F. F. Dodds suggested that the members present rise and bow their heads for a few moments in silent respect for his memory. The secretary was instructed to write to Mrs. Anderson and her son, F. Paul Anderson expressing the sympathy of the Chapter.

The postponed program of the April meeting was continued with a resumption of the discussion by Messrs. Flarsheim and Filkins of the relative merits of mechanical refrigeration and refrigeration by ice. Their presentation of the subject was interesting and was well received by the members present.

The president expressed to Mr. Campbell, Chairman of the Entertainment Committee, the thanks of the Chapter for the valuable and instructive programs which were presented during the year.

C. V. Haynes, national president of the Society, was introduced by W. A. Russell, Council Member. Mr. Haynes spoke on Your Society—What Does It Mean to You? He gave a great deal of information on the activities of the Society, from which those present derived much good and extended an invitation to attend the summer meeting at Buck Hill Falls, Pa. At the conclusion of his talk he was given a unanimous vote of thanks.

F. H. Gaylord, of the Illinois Chapter, was introduced as a guest.

No further business appearing, the meeting was adjourned.

Death of Harry A. Wilson

It was with extreme regret that word was received of the death of Harry A. Wilson of Johnston, Rhode Island on May 31, 1934.

Mr. Wilson was born on March 18, 1858 at Johnston, R. I., and his entire life was spent in the New England states. He received his preparatory education at Plainfield Academy, Plainfield, Conn., and attended Bryan-Stratton Business College. From

Harry A. Wilson

1874 to 1877 he served as steamfitters' helper and apprentice with Hale & Co., Moosup, Conn. Later he was employed as draftsman with Clarence Burlingame at Providence, R. I., and from 1889 to 1895 he was with Smith & Winchester Co. in their Providence and Boston offices where he served as draftsman and engineer. He later was connected with Smith & Thayer, consulting engineers, at Boston, Mass., as heating and ventilating engineer. Mr. Wilson retired in 1931 at which time he was with the U. S. Radiator Corp.

Mr. Wilson was a Life Member of the Society and it is interesting to note that his application for membership was sponsored by W. M. Mackay who served as President of the Society in 1897 and was Secretary from 1899 to 1910.

Funeral services were conducted by the Masonic Order of which he was a Thirty-second Degree member and burial was at Moosup, Conn. The Officers and Council of the Society feel the loss of their older members very keenly and have expressed their sorrow to Mrs. Harry A. Wilson, his widow who survives, as well as a brother, Prof. G. G. Wilson of Harvard University.

Smoke Prevention Association Elects Officers

At the Buffalo, N. Y., meeting of the *Smoke Prevention Association* the following officers were elected for the coming year: *President,* Frank Trumble; *1st Vice President,* W. E. E. Koepler; *2nd Vice President,* William Langlands; *Secretary-Treasurer,* F. A. Chambers, and *Sergeant-at-Arms,* H. V. Carlson.

Among the speakers at the meeting were: R. M. Conner, Research Director of the *American Gas Association,* Cleveland, O., whose subject was The Smoke Evil, and H. H. Toombs, chief operating engineer, Armour & Co., Chicago, Ill., who discussed the Practical Considerations for the Reduction of Smoke in Large Boiler Rooms.

Manufacturers of Unit Air Conditioners Organize

A new organization called the *Unit Air Conditioner Manufacturers Association* has recently been organized and the officers elected are as follows: *President,* P. A. McKittrick; *Vice President,* Allston Sargent; *Treasurer,* H. C. Grubbs, *Temporary Secretary,* John A. Dewhurst; *Board of Directors,* Thornton Lewis, *Chairman,* H. C. Grubbs, J. J. Donovan, J. W. Speer, J. F. G. Miller, S. E. Lauer.

Headquarters of the Association are at 330 West 42nd Street, New York City, and a code of fair trade practices has been sponsored by the organization and filed with the National Recovery Administration. The functions of the Association, which represents over 80 per cent of current production, is to foster and promote the development and the use of unit type equipment.

There are many problems of engineering that confront every proposed job so that sales organizations must be developed along proper lines to perform the required type of service. It is the aim of the Association to guide the development of the industry along the proper lines so that the advantages to be gained by the buying public will enhance the continued development and use of air conditioning.

Preliminary figures indicate that the annual gross sales for the past year of unit type equipment was approximately $4,000,000 and the current year will see this volume exceeded by 50 per cent.

Federal Better Housing Program

Community campaigns to further the modernization plans in connection with the Better Housing Program of the Federal Housing Administration have been announced by Administrator James A. Moffett. A booklet giving detailed suggestions and the methods to be used in organizing civic, business and social organizations and coordinating the efforts of financial institutions, real estate boards, merchants, supply dealers, contractors and others, has been prepared so that local communities can get the greatest benefit from the National Housing Act.

An introductory chart shows the family dwelling units built from 1921 to 1933 and a discussion of the purposes of the National Housing Act are given. The success of a few modernization campaigns is tabulated and the detailed plan for a local organization to conduct a better housing campaign is explained. The work of the various committees is specified and the extent of the field work is outlined.

The booklet concludes with essential information of (1) Who are eligible for loans; (2) Terms and conditions of the loan; (3) Procedure to obtain a loan.

The field organization of the Federal Housing Administration is directed from Washington, D. C., by W. D. Flanders and the 10 regional directors are: No. 1—Julian Gerard, 11 W. 42nd St., Rm. 1280, New York City; No. 2—Chas. A. Birmingham, 10 Post Office Square, Boston, Mass.; No. 3—Charles Edison, 505 Industrial Office Bldg., Newark, N. J.; No. 4—John W. Millsaps, 1101 First National Bank Bldg., Atlanta, Ga.; No. 5—Hugh Humphreys, 210 Old Federal Bldg., Memphis, Tenn.; No. 6—Benedict Crowell, 501 Bulkley Bldg., Cleveland, Ohio; No. 7—Percy Wilson, 134 N. La Salle St., Rm. 206, Chicago, Ill.; No. 8—Walter D. Cline, 713 R. A. Long Bldg., Kansas City, Mo.; No. 9—Gus H. Harvey, 1324 S. Minnesota Ave., Sioux Falls, S. D.; No. 10—Alfred Swinterton, 225 Bush St., San Francisco, Calif.

Standards Specified by NRA Codes

One hundred and ninety-five of the first 465 Codes and 78 Supplemental Codes approved by the National Recovery Administration contain some sort of standards clause or provision for standards, a careful analysis of each of these documents made by the *American Standards Association* shows.

Because of the great diversity of the standards provisions, and the tendency for many of these to overlap with other fair practice or fair trade provisions, an exact classification is impossible. The language used in setting forth the provisions of the individual codes differs, as must be expected as a result of the large number of business executives, Government officials, and other experts who have had a part in writing these codes, with the result that some of the "quality standards" might be put either under the head of "quality" or of "performance standards," depending upon the point of view of the analyst.

To simplify this analysis, clauses of the following types have been omitted: provisions prohibiting substitution of higher or lower qualities, when no definite grade specifications are given; provisions for product classifications for the purpose of pricing only; provisions concerning standards guarantee, on a time or other basis, when no specifications as to quality or performance are included; general prohibitions against false marking or branding, without any specific application.

In the 465 approved NRA Codes and the 78 Supplemental Codes analyzed, 44 establish specific standards of quality, performance, weight and sizes of the product, or specify standards for packaging or for containers to be used.

In addition to these, or in conjunction with them, 14 Codes have standards provisions which include either prohibition of or restrictions upon the use of seconds, etc.

Of the whole list of approved codes, 63 have provisions making reference to standards set by, or to be formed in conjunction with, some Government agency or industrial group. The specific agencies cited as authorities for standards include the National Bureau of Standards, the American Standards Association, the American Society for Testing Materials, Federal Trade Commission, U. S. Public Health Service, U. S. Department of Agriculture, and the Interstate Commerce Commission, together with the following professional societies and trade associations which have done pioneering work in establishing their own industrial standards:

American Medical Association
American Society of Heating and Ventilating Engineers
American Gas Association
American Petroleum Institute
National Board of Fire Underwriters
American Lumber Standards, or lumber manufacturers' Assn. Standards
American Railway Association
Irish Linen Industry
National Association of Glove Manufacturers
Locomotive Crane Manufacturers Association
Grinding Wheel Manufacturers Association
Tire and Rim Association
American Photo-Engravers' Association
Institute of Carpet Manufacturers
Saw Manufacturers Association
National Assn. of Flat Rolled Steel Manufacturers
National Association of Sheet Metal Contractors
National Slate Association
United Roofing Contractors Association

Of the Codes studied, 32 have standards provisions which include regulations concerning marketing, labeling, classifying, etc., other than ordinary false marketing and branding prohibitions.

Because of the dearth of acceptable standards, it was found that 93 approved NRA Codes contain provisions requiring the Code Authority in each case to take steps to formulate, or to recommend for adoption, standards to govern industry operations.